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Standards Support and Environmental Impact Statement Volume I: Proposed Standards of Performance

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EPA-450/2-78-125a

Stationary Internal Combustion Engines

Standards Support and Environmental Impact Statement Volume I: Proposed Standards of Performance

by

Emission Standards and Engineering Division

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air, Noise, and Radiation
Office of Air Quality Planning and Standards
Research Triangle Park, North Carolina 27711

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1. SUMMARY

1.1 PROPOSED STANDARDS

Standards of performance for stationary internal combustion engines are being proposed under Section 111 of the Clean Air Act. These standards would limit emissions of nitrogen oxides from diesel and dual-fuel stationary internal combustion engines with greater than 560 cubic inch displacement per cylinder (CID/cyl) and gas engines with greater than 350 cubic inch displacement per cylinder or equal to or greater than 8 cylinders and greater than 240 cubic inch displacement per cylinder.

The numerical emission limits for NO_x would be 0.0700 percent by volume (700 ppm) for gas engines, 0.0600 percent by volume (600 ppm) for diesel engines, and 0.0600 percent by volume (600 ppm) for dual-fuel engines corrected to 15 percent oxygen on a dry basis and referenced to standard atmospheric conditions of 29.92 inches mercury, 85 degrees Fahrenheit, and 75 grains moisture per pound of dry air. The proposed standards would also include an adjustment factor for engine efficiency which would adjust the emissions limits upward linearly for IC engines with thermal efficiencies greater than 35 percent. NO_x emissions from stationary internal combustion engines, therefore, would be limited according to one of the following equations:

$$STD = 700\left(\frac{5.13}{Y}\ \text{kJ/w-hr}\right) \text{for any gas engine}$$

$$STD = 600\left(\frac{5.13}{Y}\ \text{kJ/w-hr}\right) \text{for any diesel or dual-fuel engine}$$

where:

STD = allowable NO_x emissions (parts per million volume corrected to 15 percent oxygen on a dry basis).

Y = manufacturer's rated brake-specific fuel consumption at peak load (kilojoules per watt hour) or owner/operator's brake-specific fuel consumption at peak load as determined in the field.

During performance tests to determine compliance with the proposed standards, measured NO_x emissions at 15 percent oxygen would be adjusted to standard atmospheric conditions by the appropriate correction factor in the table below:

Fuel	Correction Factor
Diesel and Dual-Fuel	$K = 1/(1 + 0.00235(H - 75) + 0.00220\ (T - 85))$
Gas	$K = (K_H)\ (K_T)$ $K_H = 0.844 + 0.151\ (\frac{H}{100}) + 0.075\ (\frac{H}{100})^2$ $K_T = 1 - (T - 85)(0.0135)$

where:

H = observed humidity, grains H_2O/lb dry air

T = observed inlet air temperature, °F

Internal combustion engine manufacturers, owners, or operators may also elect to develop custom ambient condition correction factors, in terms of ambient temperature, and/or humidity, and/or ambient pressure.

These factors must also be substantiated with data and approved for use by EPA.

The proposed standards would apply to facilities that commence construction 30 months after the date of publication in the FEDERAL REGISTER.

Emergency-standby internal combustion engines and all one- and two-cylinder gas engines would be exempt from the NO_x emission limit.

1.2 ENVIRONMENTAL/ECONOMIC IMPACT

Four emission control techniques, or combinations of these techniques, have been identified as demonstrated NO_x emission reduction systems for stationary internal combustion engines. These techniques are: (1) retarded ignition or fuel injection, (2) air-to-fuel ratio changes, (3) manifold air cooling, and (4) derating power output (at constant speed). In general, all four techniques are applied by changing an engine operating adjustment.

Fuel injection retard is the most effective NO_x control technique for diesel engines. Similarly, air-to-fuel ratio change is the most effective NO_x control technique for gas engines. Both retard and air-to-fuel ratio changes are effective in reducing NO_x emissions from dual-fuel engines.

Due to technical considerations, ignition retard in excess of eight degrees in diesel engines and changes in the air-to-fuel ratio in excess of five percent in gas engines are the limits to which these techniques could be realistically applied. Eight degrees of ignition retard in diesel engines and a five percent change in air-to-fuel ratios in gas engines yield about a 40 percent reduction in NO_x emissions. Actual emissions varied among engine types; however, the degree of reduction was consistent.

Consequently, a 40 percent reduction in NO_x emissions was the most stringent regulatory option which could be selected as a basis for the standards. An alternative of 20 percent NO_x emission reduction was also considered a viable regulatory option which could serve as the basis for standards of performance.

The main environmental benefit of a standard based on either alternative would be a reduction in national NO_x emissions, which is now 14.6 million megagrams per year for all stationary sources. Total NO_x emissions would decline by 72,500 megagrams annually for alternative I (20 percent reduction) and 145,000 megagrams annually for alternative II (40 percent reduction) in the fifth year after the standard went into effect.

Ambient air quality dispersion modeling based on "worst case" conditions indicates uncontrolled ambient air NO_x levels near large stationary IC engines can vary from approximately 60 percent of the National Ambient Air Quality Standard (NAAQS) of 100 $\mu g/m^3$ to over twice the standard depending on the size of the engine. Thus, standards of performance based on alternative II would be more effective in reducing ambient air NO_x levels than standards of performance based on alternative I.

Emissions of CO and HC, however, would increase, particularly from naturally aspirated gas engines. The magnitude of the increase would be large for CO; however, for both HC and CO, significant emission reductions are readily achievable in other source categories. NO_x

emissions, on the other hand, are difficult to reduce, and stationary internal combustion engines offer one of few opportunities for significant NO_x reduction. Therefore, NO_x emissions were selected for control by standards of performance.

There would be essentially no water pollution, solid waste, or noise impact of standards of performance based on either alternative I or alternative II.

The potential energy impact of standards of performance based on either alternative is small. The potential energy impact in the fifth year after the standards go into effect, based on alternative I, would be equivalent to an increase in fuel consumption of approximately 1.0 million barrels of oil per year compared to the IC engine fuel consumption of engines affected by the standards of 31 million barrels per year. The potential energy impact in the fifth year after the standard goes into effect, based on alternative II, would be equivalent to approximately 1.5 million barrels of oil per year. The impact of alternative II represents only 0.02 percent of the 1977 domestic consumption of crude oil and natural gas and only 0.02 percent of the projected total U.S. oil imports five years after the standards go into effect.

Economic impacts on manufacturers or users of stationary internal combustion engines are small. Manufacturers of stationary internal combustion engines would incur additional costs as a result of standards of performance. The manufacturers' total capital investment requirements for developmental testing of engine models is estimated to be about $4.5 million to comply with standards of performance based on alternative I and about $5 million to comply with standards of performance based on alternative II. Analyses of the financial statements and other public financial information of engine manufacturers indicate that the manufacturers' overhead budgets are sufficient to support the development of these controls without adverse impact on their financial position. Manufacturers would not experience significant differential cost impacts among competing engine model families. Based on "worst-case" assumptions, the maximum intra-industry sales losses would be about six percent as a result of standards of performance based on either alternative. In addition, the impact with regard to increasing sales of gas turbines would be minimal, as a result of standards of performance based on either alternative.

The application of NO_x controls will also increase costs to the engine user. The magnitude of this increase will depend on the amount and type of emission control applied. Fuel penalties are the major factor affecting this increase.

A two percent increase in price would be expected on the average as the result of standards of performance based on either alternative. The total additional capital cost for all users would equal about $9.6 million on a cumulative basis over the first five years to comply with standards of performance based on either alternative. Total uncontrolled annualized costs of about $580 million by all large stationary IC engine users would increase, due mainly to fuel penalties, by about $25 million to comply with standards of performance based on alternative I and would increase by about $32 million to comply with standards of performance based on alternative II in the fifth year after the standards go into effect.

These impacts translate into price increases for the end products or services provided by the industrial and commercial users of large stationary IC engines. The electric utility industry would pass on a price increase after five years of 0.02 percent to comply with standards of performance based on either alternative. After five years, delivered natural gas prices would increase 0.02 percent as a result of standards of performance based on alternative I and 0.04 percent as a result of standards of performance based on alternative II. Even after a full phase-in period of 30 years, during which new controlled engines would replace all existing uncontrolled engines, these increases would be 0.1 percent for electric utilities, and 0.1 and 0.3 percent for delivered natural gas prices as a result of standards of performance based on alternatives I and II, respectively.

Based on this assessment of the impacts of each alternative, and since alternative II achieves a greater degree of NO_x reduction, it is selected as the best technological system of continuous emission reduction of NO_x from stationary large-bore IC engines, considering the cost of achieving such emission reduction, any nonair quality health and environmental impact, and energy requirements.

1.3 INFLATIONARY IMPACT

An economic impact analysis would have to be developed if the proposed standard caused an increase in the fifth-year annualized cost of more than $100 million, a major product price increase of five percent, or an increase in national energy consumption of 25,000 barrels of oil per day. The proposed standard of performance would increase operating costs $26 million in the fifth year, the largest price increases would be approximately two percent, and energy consumption would increase 4,300 barrels of oil per day. The Agency, therefore, feels that no economic impact analysis is required.

CHAPTER 2

INTRODUCTION

Standards of performance are proposed following a detailed investigation of air pollution control methods available to the affected industry and the impact of their costs on the industry. This document summarizes the information obtained from such a study. Its purpose is to explain in detail the background and basis of the proposed standards and to facilitate analysis of the proposed standards by interested persons, including those who may not be familiar with the many technical aspects of the industry. To obtain additional copies of this document or the Federal Register notice of proposed standards, write to EPA Library (MD-35), Research Triangle Park, North Carolina 27711. Specify Standards Support and Environmental Impact Statement: Proposed Standards of Performance for Stationary Internal Combustion Engines, report number EPA-450/3-78-125a when ordering.

2.1 AUTHORITY FOR THE STANDARDS

Standards of performance for new stationary sources are established under Section 111 of the Clean Air Act (42 U.S.C. 7411), as amended, hereafter referred to as the Act. Section 111 directs the Administrator to establish standards of performance for any category of new stationary source of air pollution which ". . . causes or contributes significantly to, air pollution which may reasonably be anticipated to endanger public health or welfare."

The Act requires that standards of performance for stationary sources reflect, ". . . the degree of emission limitation achievable through the application of the best technological system of continuous emission reduction . . . the Administrator determines has been adequately demonstrated." In addition, for stationary sources whose emissions result from fossil fuel combustion, the standard must also include a percentage reduction in emissions. The Act also provides that the cost of achieving the necessary emission reduction, the nonair quality health and environmental impacts, and the energy requirements all be taken into account in establishing standards of performance. The standards apply only to stationary sources, the construction or modification of which commences after regulations are proposed by publication in the Federal Register.

The 1977 amendments to the Act altered or added numerous provisions which apply to the process of establishing standards of performance.

1. EPA is required to list the categories of major stationary sources which have not already been listed and regulated under standards of performance. Regulations must be promulgated for these new categories on the following schedule:

   25 percent of the listed categories by August 7, 1980

   75 percent of the listed categories by August 7, 1981

   100 percent of the listed categories by August 7, 1982

   A governor of a State may apply to the Administrator to add a category which is not on the list or to revise a standard of performance.

2. EPA is required to review the standards of performance every 4 years, and if appropriate, revise them.
3. EPA is authorized to promulgate a design, equipment, work practice, or operational standard when an emission standard is not feasible.
4. The term "standards of performance" is redefined and a new term "technological system of continuous emission reduction" is defined. The new definitions clarify that the control system must be continuous and may include a low-polluting or nonpolluting process or operation.
5. The time between the proposal and promulgation of a standard under Section 111 of the Act is extended to 6 months.

Standards of performance, by themselves, do not guarantee protection of health or welfare because they are not designed to achieve any specific air quality levels. Rather, they are designed to reflect the degree of emission limitation achievable through application of the best adequately demonstrated technological system of continuous emission reduction, taking into consideration the cost of achieving such emission reduction, any nonair quality health and environmental impact and energy requirements.

Congress had several reasons for including these requirements. First, standards with a degree of uniformity are needed to avoid situations where some States may attract industries by relaxing standards relative to other States. Second, stringent standards enhance the potential for long-term growth. Third, stringent standards may help achieve long-term cost savings by avoiding the need for more expensive retrofitting when pollution ceilings may be reduced in the future.

Fourth, certain types of standards for coal burning sources can adversely affect the coal market by driving up the price of low-sulfur coal or effectively excluding certain coals from the reserve base because their untreated pollution potentials are high. Congress does not intend that new source performance standards contribute to these problems. Fifth, the standard-setting process should create incentives for improved technology.

Promulgation of standards of performance does not prevent State or local agencies from adopting more stringent emission limitations for the same sources. States are free under Section 116 of the Act to establish even more stringent emission limits than those established under Section 111 or those necessary to attain or maintain the national ambient air quality standards (NAAQS) under Section 110. Thus, new sources may in some cases be subject to limitations more stringent than standards of performance under Section 111, and prospective owners and operators of new sources should be aware of this possibility in planning for such facilities.

A similar situation may arise when a major emitting facility is to be constructed in a geographic area which falls under the prevention of significant deterioration of air quality provisions of Part C of the Act. These provisions require, among other things, that major emitting facilities to be constructed in such areas are to be subject to best available control technology. The term "best available control technology" (BACT), as defined in the Act, means ". . . an emission limitation based on the maximum degree of reduction of each pollutant subject to regulation under this Act emitted from or which results from any major emitting facility, which the permitting authority, on a case-by-case basis, taking into account energy, environmental, and economic impacts and other costs, determines is achievable for such facility through application of production processes and available methods, systems, and techniques, including fuel cleaning or treatment or innovative fuel combustion techniques for control of each such pollutant. In no event shall application of best available control technology result in emissions of any pollutants which will exceed the emissions allowed by any applicable standard established pursuant to Section 111 or 112 of this Act."

Although standards of performance are normally structured in terms of numerical emission limits where feasible, alternative approaches are sometimes necessary. In some cases physical measurement of emissions from a new source may be impractical or exorbitantly expensive. Section 111(h) provides that the Administrator may promulgate a design or equipment standard in those cases where it is not feasible to prescribe or enforce a standard of performance. For example, emissions of hydrocarbons from storage vessels for petroleum liquids are greatest during tank filling. The nature of the emissions, high concentrations for short periods during filling, and low concentrations for longer periods during storage, and the configuration of storage tanks make direct emission measurement impractical. Therefore, a more practical approach to standards of performance for storage vessels has been equipment specification.

In addition, Section 111(h) authorizes the Administrator to grant waivers of compliance to permit a source to use innovative continuous emission control technology. In order to grant the waiver, the Administrator must find: (1) a substantial likelihood that the technology will produce greater emission reductions than the standards require, or an equivalent reduction at lower economic, energy or environmental cost;

(2) the proposed system has not been adequately demonstrated; (3) the technology will not cause or contribute to an unreasonable risk to public health, welfare or safety; (4) the governor of the State where the source is located consents; and that, (5) the waiver will not prevent the attainment or maintenance of any ambient standard. A waiver may have conditions attached to assure the source will not prevent attainment of any NAAQS. Any such condition will have the force of a performance standard. Finally, waivers have definite end dates and may be terminated earlier if the conditions are not met or if the system fails to perform as expected. In such a case, the source may be given up to 3 years to meet the standards, with a mandatory progress schedule.

2.2 SELECTION OF CATEGORIES OF STATIONARY SOURCES

Section 111 of the Act directs the Administrator to list categories of stationary sources which have not been listed before. The Administrator, ". . . shall include a category of sources in such list if in his judgment it causes, or contributes significantly to, air pollution which may reasonably be anticipated to endanger public health or welfare." Proposal and promulgation of standards of performance are to follow while adhering to the schedule referred to earlier.

Since passage of the Clean Air Amendments of 1970, considerable attention has been given to the development of a system for assigning priorities to various source categories. The approach specifies areas of interest by considering the broad strategy of the Agency for implementing the Clean Air Act. Often, these "areas" are actually pollutants which are emitted by stationary sources. Source categories which emit these pollutants were then evaluated and ranked by a process involving such factors as: (1) the level of emission control (if any) already required by

State regulations; (2) estimated levels of control that might be required from standards of performance for the source category; (3) projections of growth and replacement of existing facilities for the source category; and (4) the estimated incremental amount of air pollution that could be prevented, in a preselected future year, by standards of performance for the source category. Sources for which new source performance standards were promulgated or are under development during 1977 or earlier, were selected on these criteria.

The Act amendments of August 1977, establish specific criteria to be used in determining priorities for all source categories not yet listed by EPA. These are:

1. The quality of air pollutant emissions which each such category will emit, or will be designed to emit;
2. The extent to which each such pollutant may reasonably be anticipated to endanger public health or welfare; and
3. The mobility and competitive nature of each such category of sources and the consequent need for nationally applicable new source standards of performance.

In some cases, it may not be feasible to immediately develop a standard for a source category with a high priority. This might happen when a program of research is needed to develop control techniques or because techniques for sampling and measuring emissions may require refinement. In the developing of standards, differences in the time required to complete the necessary investigation for different source categories must also be considered. For example, substantially more time may be necessary if numerous pollutants must be investigated from a single source category. Further, even late in the development process, the schedule for completion of a standard may change. For example, inability to obtain emission data from well controlled sources in time to pursue the development process in a systematic fashion may force a change in scheduling. Nevertheless, priority ranking is, and will continue to be, used to establish the order in which projects are initiated and resources assigned.

After the source category has been chosen, determining the types of facilities within the source category to which the standard will apply must be decided. A source category may have several facilities that cause air pollution, and emissions from some of these facilities may be insignificant or very expensive to control. Economic studies of the source category and of applicable control technology may show that air pollution control is better served by applying standards to the more severe pollution sources. For this reason, and because there be no adequately demonstrated system for controlling emissions from certain facilities, standards often do not apply to all facilities at a source. For the same reasons, the standards may not apply to all air pollutants emitted. Thus, although a source category may be selected to be covered by a standard of performance, not all pollutants or facilities within that source category may be covered by the standards.

2.3 PROCEDURE FOR DEVELOPMENT OF STANDARDS OF PERFORMANCE

Standards of performance must: (1) realistically reflect best demonstrated control practice; (2) adequately consider the cost, and the nonair quality health and environmental impacts and energy requirements of such control; (3) be applicable to existing sources that are modified or reconstructed as well as new installations; and (4) meet these conditions for all variations of operating conditions being considered anywhere in the country.

The objective of a program for development of standards is to identify the best technological system of continuous emission reduction which has been adequately demonstrated. The legislative history of Section 111 and various court decisions make clear that the Administrator's judgment of what is adequately demonstrated is not limited to systems that are in actual routine use. The search may include a technical assessment of control systems which have been adequately demonstrated but for which there is limited operational experience. In most cases, determination of the ". . . degree of emission reduction achievable ..." is based on results of tests of emissions from well controlled existing sources. At times, this has required the investigation and measurement of emissions from control systems found in other industrialized countries that have developed more effective systems of control than those available in the United States.

Since the best demonstrated systems of emission reduction may not be in widespread use, the data base upon which standards are developed may be somewhat limited. Test data on existing well controlled sources are obvious starting points in developing emission limits for new sources. However, since the control of existing sources generally represents retrofit technology or was originally designed to meet an existing State or local regulation, new sources may be able to meet more stringent emission standards. Accordingly, other information must be considered before a judgment can be made as to the level at which the emission standard should be set.

A process for the development of a standard has evolved which takes into account the following considerations:

1. Emissions from existing well controlled sources as measured
2. Data on emissions from such sources are assessed with consideration of such factors as: (a) how representative the tested source is in regard to feedstock, operation, size, age, etc.; (b) age and maintenance of the control equipment tested; (c) design uncertainties of control equipment being considered; and (d) the degree of uncertainty that new sources will be able to achieve similar levels of control.
3. Information from pilot and prototype installations, guarantees by vendors of control equipment, unconstructed but contracted projects, foreign technology, and published literature are also considered during the standard development process. This is especially important for sources where "emerging" technology appears to be a significant alternative.
4. Where possible, standards are developed which permit the use of more than one control technique or licensed process.
5. Where possible, standards are developed to encourage or permit the use of process modifications or new processes as a method of control rather than "add-on" systems of air pollution control.
6. In appropriate cases, standards are developed to permit the use of systems capable of controlling more than one pollutant. As an example, a scrubber can remove both gaseous and particulate emissions, but an electrostatic precipitator is specific to particulate matter.

7. Where appropriate, standards for visible emissions are developed in conjunction with concentration/mass emission standards. The opacity standard is established at a level that will require proper operation and maintenance of the emission control system installed to meet the concentration/mass standard on a day-to-day basis. In some cases, however, it is not possible to develop concentration/mass standards, such as with fugitive sources of emissions. In these cases, only opacity standards may be developed to limit emissions.

2.4 CONSIDERATION OF COSTS

Section 317 of the Act requires, among other things, an economic impact assessment with respect to any standard of performance established under Section 111 of the Act. The assessment is required to contain an analysis of:

(1) the costs of compliance with the regulation and standard including the extent to which the cost of compliance varies depending on the effective date of the standard or regulation and the development of less expensive or more efficient methods of compliance;

(2) the potential inflationary recessionary effects of the standard or regulation;

(3) the effects on competition of the standard or regulation with respect to small business;

(4) the effects of the standard or regulation on consumer cost, and,

(5) the effects of the standard or regulation on energy use.

Section 317 requires that the economic impact assessment be as extensive as practicable, taking into account the time and resources available to EPA.

The economic impact of a proposed standard upon an industry is usually addressed both in absolute terms and by comparison with the control costs that would be incurred as a result of compliance with typical existing State control regulations. An incremental approach is taken since both new and existing plants would be required to comply with State regulations in the absence of a Federal standard of performance. This approach requires a detailed analysis of the impact upon the industry resulting from the cost differential that exists between a standard of performance and the typical State standard.

The costs for control of air pollutants are not the only costs considered. Total environmental costs for control of water pollutants as well as air pollutants are analyzed wherever possible.

A thorough study of the profitability and price-setting mechanisms of the industry is essential to the analysis so that an accurate estimate of potential adverse economic impacts can be made. It is also essential to know the capital requirements placed on plants in the absence of Federal standards of performance so that the additional capital requirements necessitated by these standards can be placed in the proper perspective. Finally, it is necessary to recognize any constraints on capital availability within an industry, as this factor also influences the ability of new plants to generate the capital required for installation of additional control equipment needed to meet the standards of performance.

2.5 CONSIDERATION OF ENVIRONMENTAL IMPACTS

Section 102(2)(C) of the National Environmental Policy Act (NEPA) of 1969 requires Federal agencies to prepare detailed environmental impact statements on proposals for legislation and other major Federal actions significantly affecting the quality of the human environment. The objective of NEPA is to build into the decision-making process of Federal agencies a careful consideration of all environmental aspects of proposed actions.

In a number of legal challenges to standards of performance for various industries, the Federal Courts of Appeals have held that environmental impact statements need not be prepared by the Agency for proposed actions under Section 111 of the Clean Air Act. Essentially, the Federal Courts of Appeals have determined that ". . . the best system of emission reduction, . . . require(s) the Administrator to take into account counter-productive environmental effects of a proposed standard, as well as economic costs to the industry . . ." On this basis, therefore, the Courts ". . . established a narrow exemption from NEPA for EPA determination under Section 111."

In addition to these judicial determinations, the Energy Supply and Environmental Coordination Act (ESECA) of 1974 (PL-93-319) specifically exempted proposed actions under the Clean Air Act from NEPA requirements. According to Section 7(c)(1), "No action taken under the Clean Air Act shall be deemed a major Federal action significantly affecting the quality of the human environment within the meaning of the National Environmental Policy Act of 1969."

The Agency has concluded, however, that the preparation of environmental impact statements could have beneficial effects on certain regulatory actions. Consequently, while not legally required to do so by Section 102(2)(C) of NEPA, environmental impact statements will be prepared for various regulatory actions, including standards of performance developed under Section 111 of the Act. This voluntary preparation of environmental impact statements, however, in no way legally subjects the Agency to NEPA requirements.

To implement this policy, a separate section is incuded in this document which is devoted solely to an analysis of the potential environmental impacts associated with the proposed standards. Both adverse and beneficial impacts in such areas as air and water pollution, increased solid waste disposal, and increased energy consumption are identified and discussed.

2.6 IMPACT ON EXISTING SOURCES

Section 111 of the Act defines a new source as ". . . any stationary source, the construction or modification of which is commenced . . ." after the proposed standards are published. An existing source becomes a new source if the source is modified or is reconstructed. Both modification and reconstruction are defined in amendments to the general provisions of Subpart A of 40 CFR Part 60 which were promulgated in the Federal Register on December 16, 1975 (40 FR 58416). Any physical or operational change to an existing facility which results in an increase in the emission rate of any pollutant for which a standard applies is considered a modification. Reconstruction, on the other hand, means the replacement of components of an existing facility to the extent that the fixed capital cost exceeds 50 percent of the cost of constructing a comparable entirely new source and that it be technically and economically feasible to meet the applicable standards. In such cases, reconstruction is equivalent to new construction.

Promulgation of a standard of performance requires States to establish standards of performance for existing sources in the same industry under Section 111(d) of the Act if the standard for new sources limits emissions of a designated pollutant (i.e., a pollutant for which air quality criteria have not been issued under Section 108 or which has not been listed as a hazardous pollutant under Section 112). If a State does not act, EPA must establish such standards. General provisions outlining procedures for control of existing sources under Section 111(d) were promulgated on November 17, 1975, as Subpart B of 40 CFR Part 60 (40 FR 53340).

2.7 REVISION OF STANDARDS OF PERFORMANCE

Congress was aware that the level of air pollution control achievable by any industry may improve with technological advances. Accordingly, Section 111 of the act provides that the Administrator ". . . shall, at least every 4 years, review and, if appropriate, revise . . ." the standards. Revisions are made to assure that the standards continue to reflect the best systems that become available in the future. Such revisions will not be retroactive but will apply to stationary sources constructed or modified after the proposal of the revised standards.

CHAPTER 3

THE STATIONARY RECIPROCATING INTERNAL COMBUSTION ENGINE PROCESS AND INDUSTRY

3.1 GENERAL

Stationary reciprocating internal combustion (IC) engines operate on the same principles as the common automobile or truck engine. They can be installed almost anywhere, since they can be instrumented for remote operation, can use gasoline, diesel fuel, natural gas, sewage gas, and certain mixtures of these fuels, and require relatively little water. Engines are manufactured in sizes ranging from less than 1 hp to nearly 50,000 hp, although the largest one built in the United States is 13,500 hp. Installations characteristically have a low physical profile (low buildings, short stacks, little visible emissions, and quiet operation when properly muffled), at least when installed as single units. They are often located in, or adjacent to, large urban centers, where power demands are greatest and pollution problems are often the most severe.

IC engines are being used in a multitude of applications because of their short construction time, ease of installation, and remote operation capability with a variety of fuels over a large range of speeds and loads. These applications range from driving large municipal electrical generators to powering small air compressors and welders. Further details about the manufacturers and users of IC engines are presented in the following subsections.

3.1.1 Engine Manufacturers

Approximately 40 firms manufacture IC engines for stationary applications in the United States, and many others produce forgings, fuel systems, turbochargers, heat exchangers, and related components. In addition, many Original Equipment Manufacturers (OEM's) purchase engines to incorporate into such final products as trucks, tractors, compressors, welders, pumps, generator sets, and other machinery. These OEM's are significant customers for some manufacturers, particularly those who make truck and tractor engines. One such manufacturer has stated that he sells to about 400 OEM's, whereas another claims his company deals with 40[1]. In addition, several of the manufacturers of small- and medium-sized engines sell their product to dealers and distributors, who, in turn, sell them to the end users. Therefore, the manufacturing companies have little knowledge or control over the end-use of their engines. To further complicate the situation, some engine manufacturers are themselves OEM's; they buy engines from other manufacturers in size ranges that they do not produce, and then mate these purchased engines with their own compressors, generators, etc.

For the purposes of this discussion, manufacturers of IC engines can be categorized into four major groups: (1) firms manufacturing large-bore (greater than 8 inches), low- and medium-speed (less than 1200 rpm) engines; (2) firms manufacturing principally small-bore (less than 6-1/2 inches), medium-power, high-speed engines (greater than 1200 rpm); (3) firms manufacturing low-power (less than 100 hp), high-speed engines and generator sets; and (4) firms which manufacture small (less than 20 hp) one-cylinder, air-cooled gasoline engines principally used for lawn and garden equipment. Table 3-1[2] shows these firms in their respective categories. Although there is some overlap in power ratings offered by members of the first three

TABLE 3-1. IC ENGINE MANUFACTURERS(2)

T-660

Manufacturer	Bore, inches	CID/cyl[a]	Power Range, hp[b]	Cylinder Power, hp/cyl	Speed Range, rpm
Large Bore Engines					
Alco	9	666	1000 - 4400	175 - 250	400-1200
Colt	8 - 16	1037-3526	850 - 9400	175 - 500	500-900
Cooper-Bessemer	13 - 20	2155-6283	900 - 13500	360 - 675	250-600
ElectroMotive Div (GM)	9	645	800 - 3600	100 - 180	900
Enterprise	13 - 17	1200-4770	1600 - 13500	280 - 680	400-600
Dresser-Clark	17 - 19	3860-5100	1000 - 10000	200 - 500	300-330
Ingersoll-Rand	11 - 17	1350-4993	1000 - 5500	125 - 330	300-550
Worthington[c]	14 - 16	2972-4021	2300 - 8600	125 - 330	300-500
White Superior	8 - 14	510-825	400 - 2650	75 - 150	900-1000
Medium Bore Engines					
Allis-Chalmers	3-1/2 - 6	43-169	29 - 850	10 - 70	1200-2600
Case	3-3/4 - 4-1/8	47-84	50 - 125	12 - 30	1200-2200
Caterpillar	4-1/2 - 6-1/4	79-246	85 - 1000	21 - 63	1200-2400
Chevrolet-Oldsmobile (GM)	3-7/8 - 4-1/4	26-57	50 - 215	13 - 27	3600-4000
Chrysler	3-1/4 - 4-1/4	28-52	43 - 175	12 - 30	1200-4000
Cooper-Penjax[d]	5 - 15	128-2827	15 - 300	15 - 150	200-900
Cummins	4-1/2 - 6-1/4	63-192	120 - 1200	20 - 100	1200-2100
Detroit Diesel	3-7/8 - 5-3/4	53-149	50 - 1100	20 - 68	1800-2500
Ford	4-1/8 - 4-1/2	26-67	38 - 200	9 - 22	2500-4600
International Harvester	3-7/8 - 5-3/8	39-136	16 - 325	9 - 50	1800-3000
John Deere	4 - 4-3/4	55-89	44 - 180	15 - 30	1500-2500
Minneapolis-Moline	4-1/4 - 5-5/16	71-133	90 - 180	15 - 29	1800
Murphy	5-1/2 - 6-3/8	142-207	110 - 520	30 - 60	1200-1800
Sterling	3 - 5-1/2	16-166	16 - 152	8 - 20	1200-1800
Stewart and Stevenson	3-7/8 - 4-3/4	53-149	30 - 1100	20 - 68	1200-1800
Teledyne Continental	3-7/8	28-41	50 - 80	12 - 14	2000-2400
Waukesha[e]	3-5/8 - 9-3/8	38-586	52 - 1550	8 - 110	1000-1800
White Engines (Hercules)	3-3/4 - 4-1/2	35-80	34 - 130	12 - 26	2400-2800

[a]Cubic inch displacement per cylinder
[b]Horsepower is for rated conditions (continuous operation) @ 130°F intercooler water temperature for large-bore engines and 85°F inlet air temperature for medium-bore engines.
[c]Worthington ceased producing engines during the writing of this report.
[d]This manufacturer produces high power one- and two-cylinder engines.
[e]The manufacturer straddles the medium- to large-bore categories, however, the majority of engine production is in the medium-bore category.

TABLE 3-1. Concluded

Manufacturer	Bore, inches	CID/cyl[a]	Power Range, hp[b]	Cylinder Power, hp/cyl	Speed Range, rpm
Small Engines and Generator Sets					
Kohler	—		3-3/4 - 28	3 - 7	2000-2700
Onan	3-1/4		8 - 30	6 - 8	1800-3900
Teledyne Wisconsin	2-3/4 - 4-1/8		3-1/2 - 80	3-1/2 - 20	2400-3600
Wills Industries	—		11 - 28	11 - 14	3000
Witte	4-1/4 - 5		9 - 27	9 - 14	800-1800
Very Small Engines					
Briggs and Stratton	2-3/8 - 3-9/16		2 - 16	2 - 16	3100-4000
Chrysler	2 - 2-1/4		3-1/4 - 8	3-1/4 - 8	5500-7000
Clinton	2-1/2 - 2-3/8		4 - 7	4 - 7	4600-5800
Homelite	1-7/16 - 2-3/4		2 - 4	2 - 4	—
Jacobsen	2-1/8		3	3	3600
Outboard Marine Corp. (Lawn Boy)	2-3/8		<10	—	—
McCulloch	1-3/8 - 2-1/4		<20	—	—
O&R	1-1/8 - 1-17/32		1 - 2-1/4	1 - 2-1/2	6300-7200
Tecumseh	2-3/8 - 3-1/2		2-1/2 - 16	2-1/2 - 16	2500-3600

[a] Cubic inch displacement per cylinder
[b] Horsepower is for rated conditions (continuous operation) @ 130°F intercooler water temperature for large-bore engines and 85°F inlet air temperature for medium-bore engines.

groups (Figure 3-1), the differences tend to be more distinct when viewed on a power-per-cylinder, displacement-per-cylinder, or bore basis (Figures 3-2, 3-3, and 3-4). The latter two show the clearest division between medium- and high-power engines, with only Waukesha and Cooper-Penjax straddling the two categories. More significant, however, is the fact that the market interactions between firms in the different groups are of far less importance than the interactions within the groups. Thus, the stationary IC engine industry is in many ways four industries, each with its own markets and problems.

Any IC engine can be adapted for use in stationary applications (for example, automotive engines are easily adapted for irrigation pumps). Thus, any engine manufacturer is a potential source of stationary engines. However, the following discussion will concentrate only on those manufacturers who market engines specifically for stationary applications.

All but three of the manufacturers are divisions or subsidiaries of large, diversified corporations with stationary engine sales accounting for a small part of total corporate revenue.[1/] Total sales of stationary engines are in the range of $300 million to $400 million per year. Since most firms manufacture engines for nonstationary as well as stationary applications, it is difficult to determine employment directly attributable to stationary engines. Assuming, however, that the percentage of employment at a plant attributable to stationary engines is equal to the percentage of sales attributable to stationary engines, it is estimated that between 15,000

[1/]The discussion in this paragraph excludes manufacturers of engines under 20 hp. Their annual sales are also in the range of $300 million to $500 million.

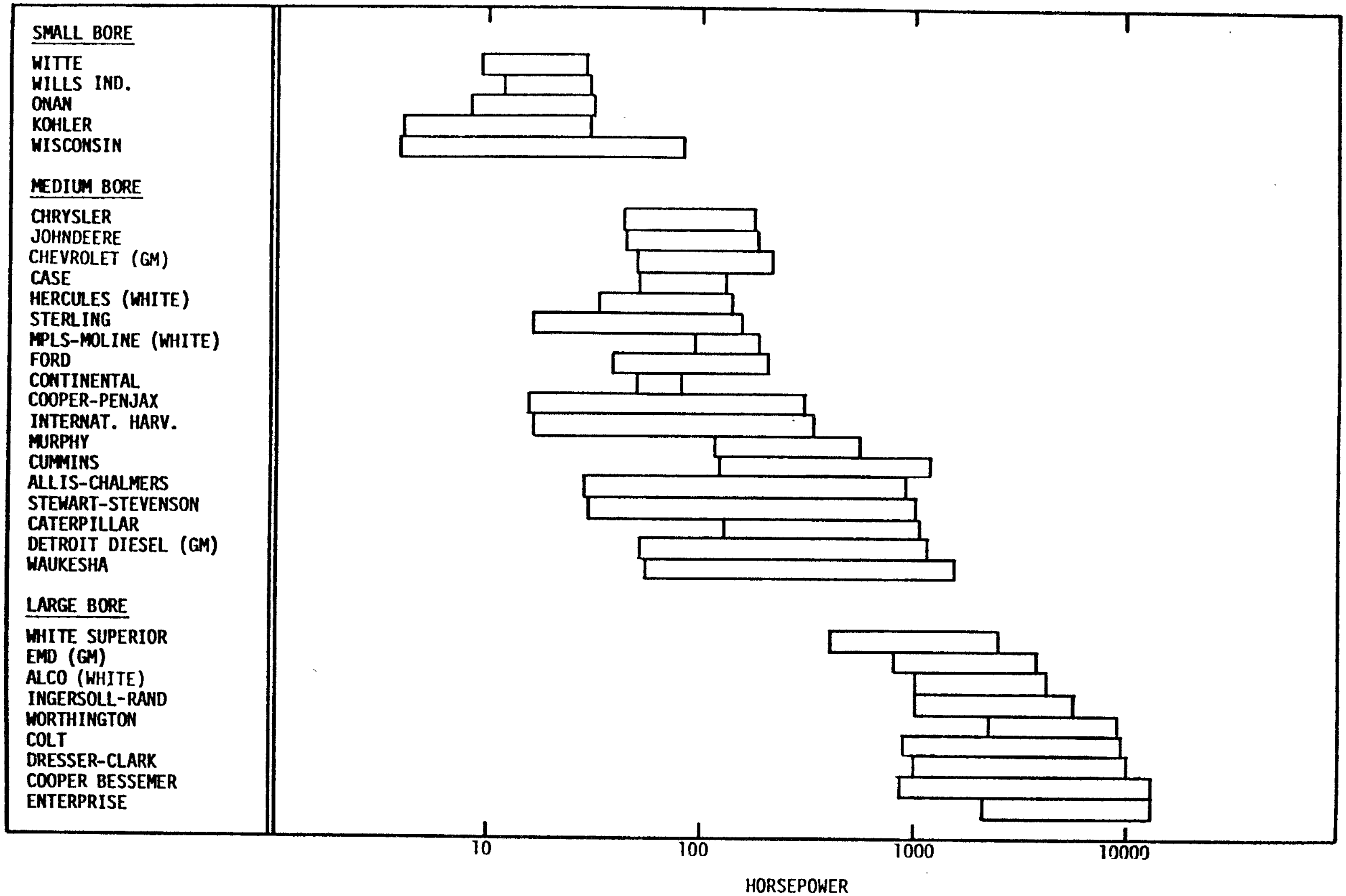


Figure 3-1. Manufacturers categorized by engine horsepower.

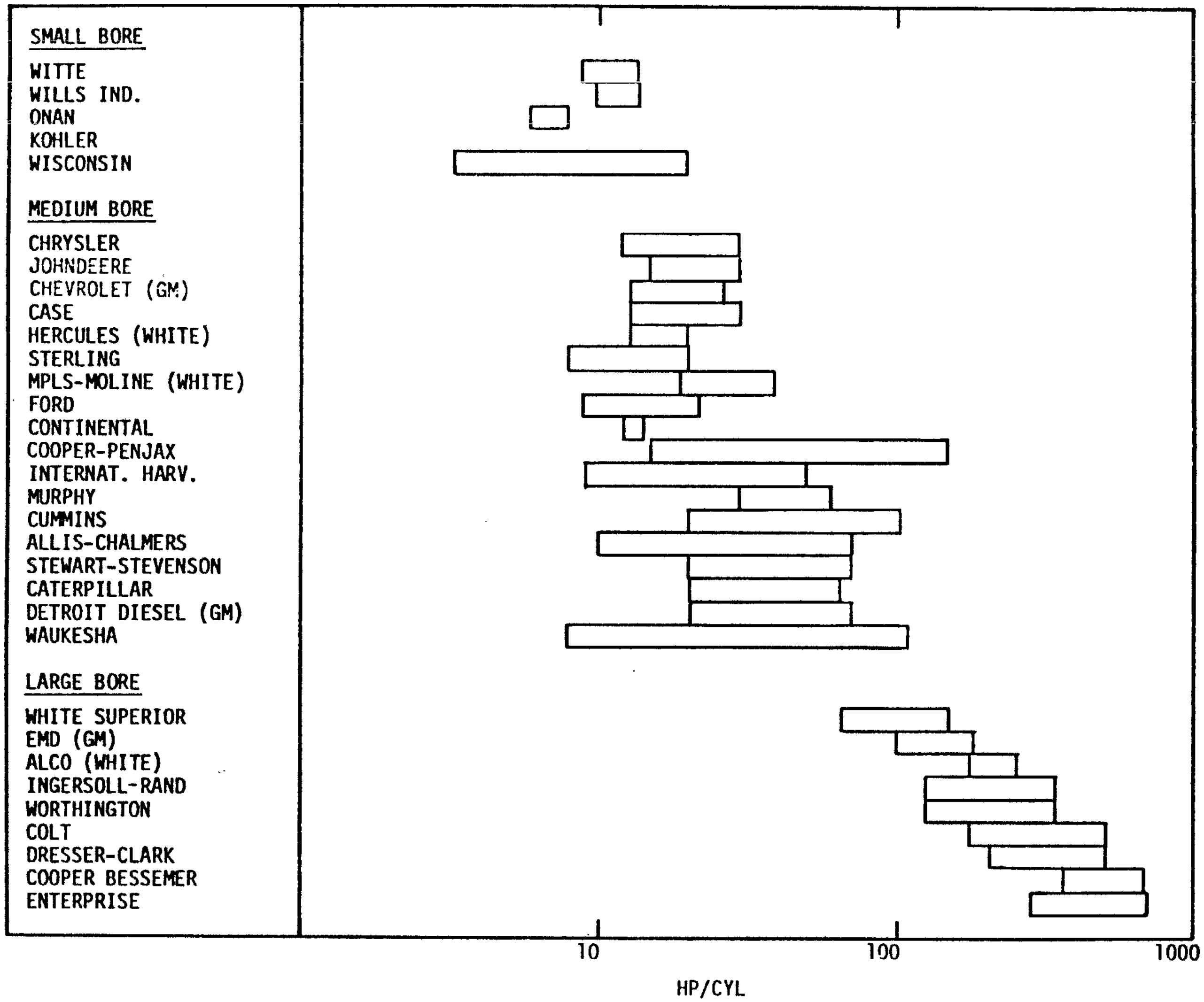


Figure 3-2. Manufacturers categorized by engine horsepower per cylinder.

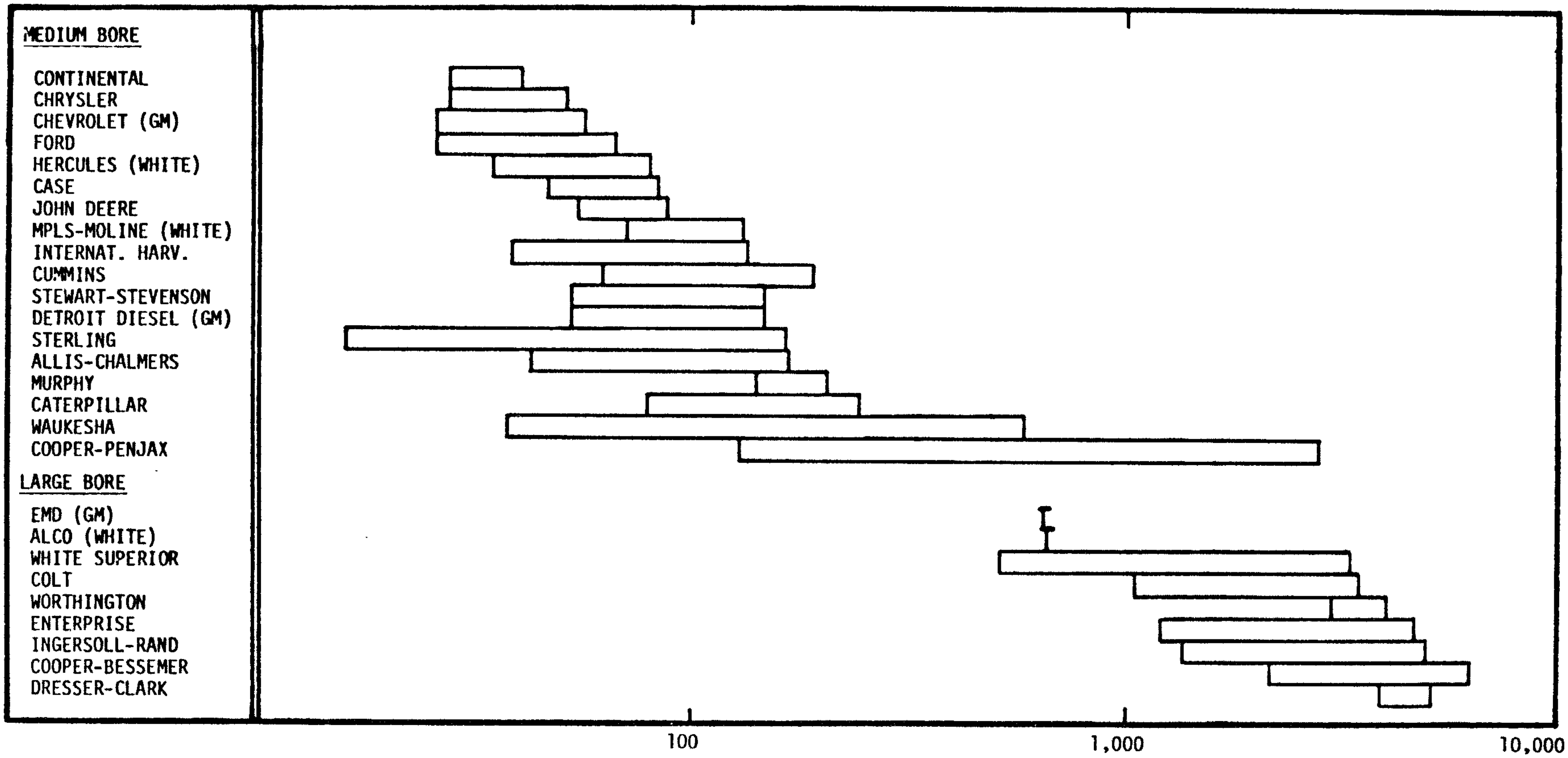


Figure 3-3. Manufacturers of medium and large engines categorized by cylinder displacement, in^3/cyl.

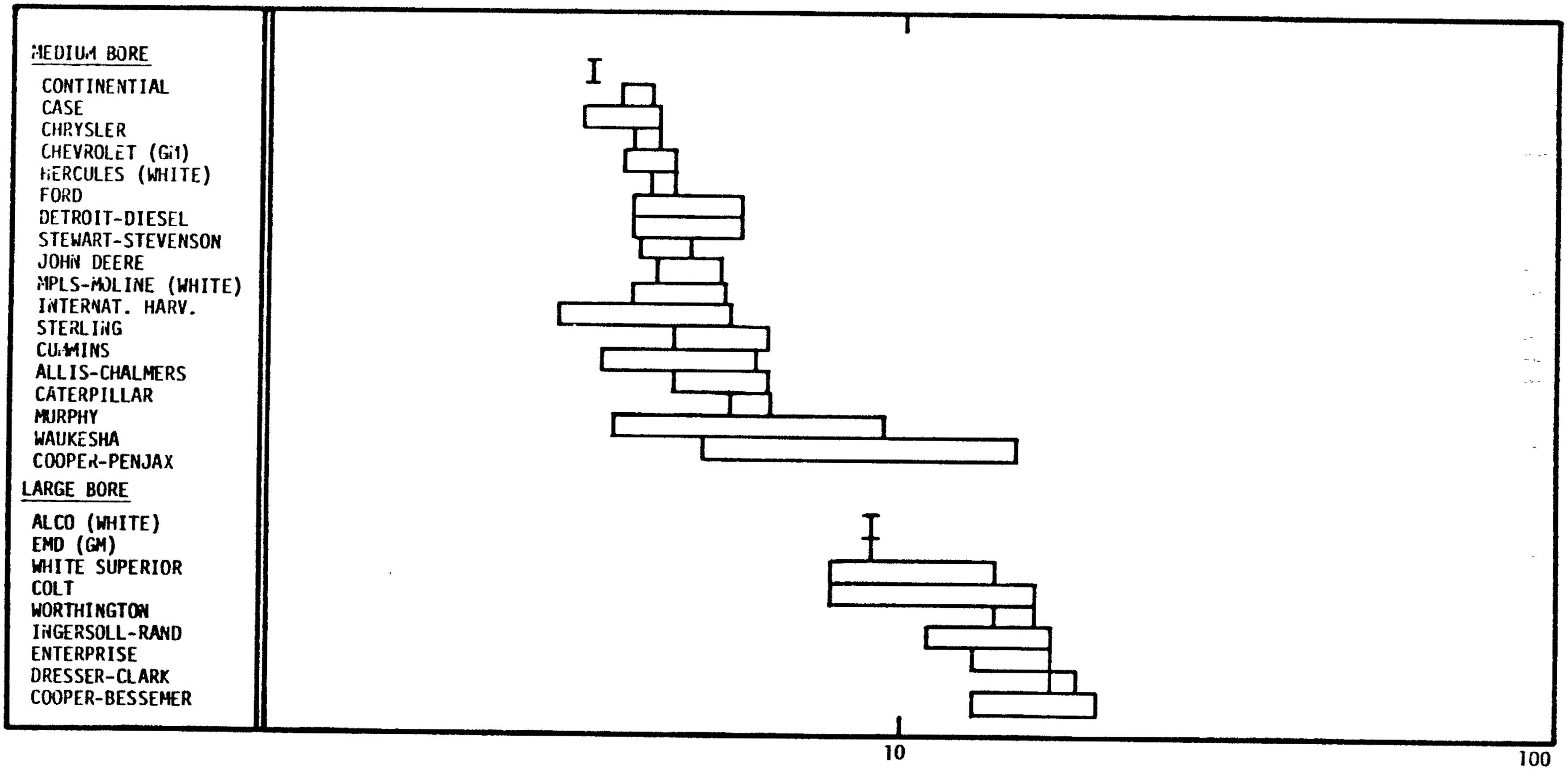


Figure 3-4. Manufacturers of medium and large engines categorized by bore size, in.

and 30,000 people are employed directly in the manufacture of stationary engines.

3.1.1.1 Large-Bore, High-Power, Low- and Medium-Speed Engines

Most of the large-bore, high-power (>100 hp/cylinder) engines manufactured are four-cycle, compression-ignition engines designed to operate on either diesel oil or a mixture of oil and natural gas (dual fuel). The remainder of the engines are spark-ignited natural gas engines, about equally distributed between two- and four-cycle varieties. Some engines are built to operate as either spark-ignited or compression-ignited (dual fuel or oil) and can easily be switched in the field in response to fuel availability. Production of large-bore engines for stationary applications is currently in the range of 1000 to 2000 units/year with a total production value of $80 million to $150 million. Sales have been largely constant over the last 5 years, with some firms showing a slight growth and others, particularly those manufacturing gas engines, having declining sales. These manufacturers expect sales to grow at an annual rate of 5 to 10 percent, on a total horsepower basis, for the next 5 years as demand increases for engines for nuclear standby and natural gas pipelines (see Section 7.1.2). Between 5000 and 6000 people are employed in the manufacture of stationary high-power engines.

As Figure 3-1 indicates, there is a region of overlap between the power ranges offered by the manufacturers of large- and medium-bore engines. There are basic differences, however, which separate the two groups. Large-bore engines produce high-power output at low speeds due to their large displacement and consequent high power per cylinder. Smaller bore engines, in contrast, have lower power per cylinder (and therefore more cylinders for the same engine horsepower) and achieve high outputs by utilizing high rotative speeds. Thus, for the same power rating, high-speed engines are smaller, less expensive, and capable of running at a wider range of speeds. Low-speed, large-bore engines, on the other hand, are more economical to operate continuously because of lower fuel consumption and longer lifetimes. For these reasons, high-speed engines tend to find different applications (see below) than low-speed engines of the same power, and there is little competition between firms in these two groups.

3.1.1.2 Medium-Power, High-Speed Engines

The firms manufacturing medium-size, high-speed engines (110 to 100 hp/cylinder and greater than 1000 rpm) vary greatly in size, employing from less than 200 to more than 10,000 people in engine manufacturing. Although four firms, Cummins, Caterpillar, Detroit Diesel Allison Division of General Motors, and Ford, dominate diesel engine production (with annual sales of 60,000 to 120,000 units each), they concentrate their sales in the truck, tractor, construction, and material handling markets(3). Ford, General Motors, and International Harvester also manufacture significant quantities of gasoline-fueled truck and tractor engines. Therefore, most of the stationary engines produced by these manufacturers are mobile engines modified for stationary installation and constant speed operation. Three additional firms, International Harvester, Waukesha, and Teledyne Continental, each sell about as many engines for stationary applications (5000 to 15,000 per year) as do the four larger companies(4,5,6). The remaining manufacturers in this group are smaller.

Medium-power engines are predominantly gasoline- and diesel-fueled; natural gas engines are less important. Because of the wide range of power offered, from 50 to 1500 hp, these engines are used in a wide variety of miscellaneous industrial, commercial, nonpropulsive marine, and agricultural applications where shaft power is needed and electricity is either unavailable or inappropriate (see Chapter 3.1.2).

Although precise production data are unavailable, we estimate that sales of diesel medium-power, high-speed engines for stationary applications have been in the range of 60,000 to 80,000 units/year over the past 6 years with a total value of $150 to $200 million per year (FOB plant), and annual sales for gasoline medium-power, high-speed engines have been approximately 100,000 with a total value of $50 million (FOB plant). Sales of these engines have been erratic from one year to the next, but the general trend seems to be upward. The industry expected this growth to continue at about the same range as the economy (that is, at 3 to 5 percent per year) before the recent downturn and concomitant economic uncertainties[7,8].

3.1.1.3 Small Engines

Five firms are classified as manufacturers of small engines and generator sets. These manufacturers are distinguished from the previously discussed manufacturers in that they produce mostly one- and two-cylinder engines of less than 50 horsepower. Although there is some overlap between the power ranges offered by the manufacturers of medium-power, high-speed engines and those of small engines, most of the engines sold by the former group are larger than 50 horsepower.

The engines produced by the five manufacturers in this group are mostly diesel and gasoline, one- and two-cylinder models. The firms also produce some four-cylinder models. All have four strokes per power cycle, and four of the firms produce air-cooled engines. Onan, Kohler, and Wills produce mainly engine-generator sets -- small, semiportable integral engines and generators that are used to provide electrical power in remote locations. Other applications for the engines include power for small pumps and blowers and off-the-road vehicles. A particularly important application for some manufacturers is refrigeration compressors for trucks and railroad cars, and hydraulic pumps for tractor-trailer dump trucks and trash compactors. Sales of small engines are in the range of $75 to $100 million per year, and employment is approximately 4000 to 6000.

3.1.1.4 Very Small Engine Manufacturers

In addition to the manufacturers discussed above, there are at least 10 firms which produce one-cylinder, air-cooled gasoline engines rated at less than 20 hp. These firms sell about 12.5 million engines per year (1973) with a total product value of $400 million(9). Therefore, the unit wholesale cost of each engine is about $30. These engines are used primarily for lawn and garden equipment and chain saws, and to a lesser degree for recreational vehicles, such as snowmobiles and small all-terrain vehicles. The dollar sales of these engines constitutes 60 percent of the value of all gasoline engines (except automotive, outboard, truck, bus, tank, and aircraft) and is approximately equal to the dollar sales of all other stationary engines (estimated at $300 million to $400 million annually, as mentioned above).

3.1.2 Engine Users

On the basis of installed horsepower, the principal stationary applications of IC engines are the following: oil and gas pipelines, oil and gas production, general industrial (including construction), electrical power generation, and agriculture. Table 3-2(10-32) shows estimated IC engine applications and usage patterns by fuel.

TABLE 3-2. STATIONARY ENGINE APPLICATIONS

Category	Annual Production, units/yr	Population,[a] units	Average Power, hp	Load Factor	Annual Usage, hr/yr	Annual Energy Production — Yearly Sales, hp-hr/yr x 10⁶	Annual Energy Production — Installed Units, hp-hr/yr x 10⁶	Constant Load	Variable Load	Constant Speed	Variable Speed	Basis of Estimate[b]
DIESEL												
Oil & Gas Production												
Offshore drilling	800[c]	675	350	0.8	2000		378		X		X	AGA Market Study
Land drilling		3,050	350	0.8	2000		1,708		X		X	AGA Market Study
Oil & Gas Transport	50[c]	500	2000	0.8	6000		4,800	X			X	McGowin, Gas Facts
Electric Generation	150[c]	400[c]	2500	0.8	2600		2,160		X	X		FPC, Diesel & Gas Power Costs
General Industrial & Agriculture												
Municipal water supply	200[c]	2,100	120	0.75	3000		567	X		X		AGA Market Study
Marine nonpropulsive	5,000[d]	15,000	100	0.5	3500	62.5	1,875		X		X	Current Industrial Report, Industry contacts
Construction, small	5,000	50,000	50	0.5	500	62.5	625		X		X	
Miscellaneous, large[e]	1,700		750	0.5	100	56.3	1,125					
Construction, large	5,000	50,000	240	0.5	500	300.0	3,000		X		X	
Portable compressors[f]	9,000	90,000	75	0.5	500	168.8	1,688		X		X	
Welders	8,000	80,000	55	0.5	500	110	1,000	X	X		X	
Pumps	5,000	25,000	100	0.5	1000	250	1,250	X		X		
Generator sets (stand by)												
<50 kW	7,000	70,000	75	0.5	500	35.0	350		X	X		
50 kW — 400 kW	8,000	160,000	250	0.5	250	250	5,000	X	X	X		
400 kW — 1000 kW	1,500	30,000	750	0.5	100	56.3	1,125	X	X	X		

Footnotes appear at end

TABLE 3-2. Continued

Category	Annual Production, units/yr	Population,[a] units	Average Power, hp	Load Factor	Annual Usage, hr/yr	Annual Energy Production – Yearly Sales, hp-hr/yr x 10^6	Annual Energy Production – Installed Units, hp-hr/yr x 10^6	Constant Load	Variable Load	Constant Speed	Variable Speed	Basis of Estimate[b]
DUAL FUEL												
Oil & Gas Transport	included in Diesel	included in Diesel		0.8	6000		2,228	X			X	McGowin, Gas Facts
Electric Generation				0.8			6,000		X	X		FPC, Diesel & Gas Power Costs
NATURAL GAS[g]												
Agriculture		91,000	100	0.75	2500		17,063	X		X		AGA Market Study
Oil & Gas Production												
Oil & gas well pumps		266,000	15	0.7	3500		9,776	X		X		AGA Market Study
Secondary recovery		5,600	200	0.8	6000		5,376	X		X		AGA Market Study
Well drilling		3,050	350	0.8	2000		1,708	X		X		AGA Market Study
Plant processing		4,000	750	0.8	8000		19,200	X		X		McGowin
Oil & Gas Industry												
Utility compressors[h]		4,500	2000	0.9	6000		51,800	X		X		Southwest Research Institute[i]
		4,000	750	0.8	6000		14,400	X		X		Southwest Research Institute[i]
Electric Generation												
Private/public utility				0.8			166.5		X	X		FPC, Diesel & Gas Power Costs
Commercial-institutional		450	200	0.45	4000		162		X	X		AGA Market Study
Stand-by		2,000	100	0.9	50		9	X		X		AGA Market Study
Industrial on-site		1,500	300	0.6	4000		1,080		X	X		AGA Market Study

Footnotes appear at end

TABLE 3-2. Continued

Category	Annual Production, units/yr	Population,[a] units	Average Power, hp	Load Factor	Annual Usage, hr/yr	Annual Energy Production — Yearly Sales, hp-hr/yr x 10^6	Annual Energy Production — Installed Units, hp-hr/yr x 10^6	Constant Load	Variable Load	Constant Speed	Variable Speed	Basis of Estimate[b]
General Industrial												
Industrial shaft power		2,900	200	0.75	5000		2,175		X		X	AGA Market Study
Plant air		750	100	0.5	4000		150.4		X		X	AGA Market Study
Air-conditioning		3,760	80	0.4	2000		240.6		X		X	AGA Market Study
Commerical shaft power		600	2000	0.6	1000		720		X		X	AGA Market Study
Municipal												
water supply		2,100	120	0.75	3000		567	X		X		AGA Market Study
waste treatment		1,740	400	0.45	4000		1,566	X		X		AGA Market Study
GASOLINE												
Agriculture												
Misc. machinery[j]	20,000	400,000	30	0.5	200	60	1,200	X	X	X	X	Current Industrial Reports, Industry contacts
Irrigation	5,000	10,000	100	0.75	2000	750	1,500	X	X	X		
General Industrial												
Generator sets >5 kW	35,000	350,000	55	0.5	400				X	X		
Compressors	7,000	70,000	55	0.5	400	715	7,150		X		X	
Welders	18,000	180,000	55	0.5	400							
Miscellaneous	5,000	50,000	55	0.5	400			X	X	X	X	
Construction	5,000	40,000	150	0.5	500	188	1,500		X		X	
Small (<15 hp)	12,600,000[k]	63,000,000	4.2	0.5	50	1100	6,615	X	X	X	X	Current Industrial Reports

See footnotes on following page.

TABLE 3-2. Concluded

Footnotes

[a]Annual production multiplied by life in years (based on estimated service life of 5000 hours for diesel engines, 4000 hours for gasoline engines, or as noted) to compute population
[b]See References 10 through 32 for complete titles of these sources.
[c]Approximated, based on estimated population and annual usage
[d]7500-hour service life assumed
[e]Applications include pumps, snow blowers, aircraft turbine starters, etc.
[f]Excludes mobile refrigeration units
[g]Population estimates come from the AGA market study. Annual production is not estimated for this category since production has been changing rapidly, decreasing continuously since 1966 (see Figure 8-3) and, therefore, an annual estimate of production would be misleading.
[h]Includes transport, distribution, gathering, and storage
[i]Reference 24
[j]Pull combines, balers, sprayers, dusters, etc.
[k]Estimated service life of 5 years

Pipeline installations are concentrated in the oil and gas producing states on the Gulf Coast and in the Midwest as shown in Figure 3-5(a)(33). Electric generation includes base load generation, principally for municipalities in the plains and Midwest, emergency standby for vital public services and large buildings in urban areas, and remote generation (for mines and homes) in rural areas. Agriculture applications are primarily irrigation pumping, concentrated in those areas with irrigated farm lands, as shown in Figure 3-5(b)(34). Additional agricultural applications include frost control, harvesting (auxiliary engines), and some remote electric generation. Construction applications include portable compressors, welders, pumps, electric generation, and material handling equipment.

For applications other than electric power generation, IC engines often compete with electric motors. Engines involve a higher initial investment but have an advantage in locations where gas and fuel oil are less expensive than electricity, such as remote or temporary construction sites. Thus, engines tend to be concentrated in areas where electric power generation and transmission costs are high, as in the mountain and prairie states, or where fuel costs are low, as in the gas-producing areas along the Gulf Coast.

3.2 PROCESSING AND THEIR EMISSIONS

3.2.1 Fundamental Description of IC Engines[2/]

Reciprocating internal combustion engines produce shaft power by confining a combustible mixture in a small volume between the head of a piston and its surrounding cylinder, causing this mixture to burn, and

[2/] Much of the material in this section is drawn from References 35 and 36. These books describe the operation of IC engines in considerable detail.

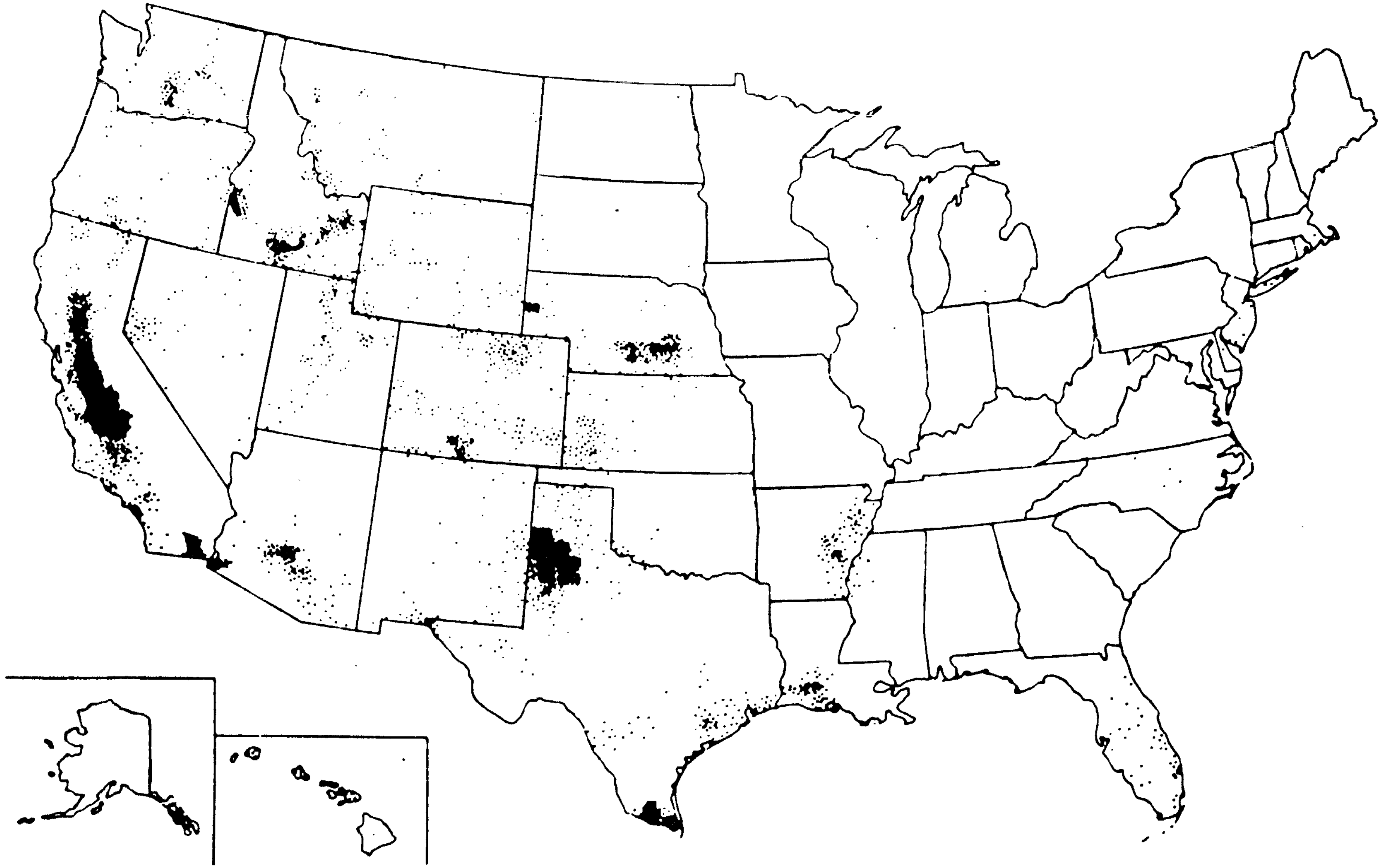


Figure 3-5(a). Potential users of stationary reciprocating engines: irrigated farms.[33]

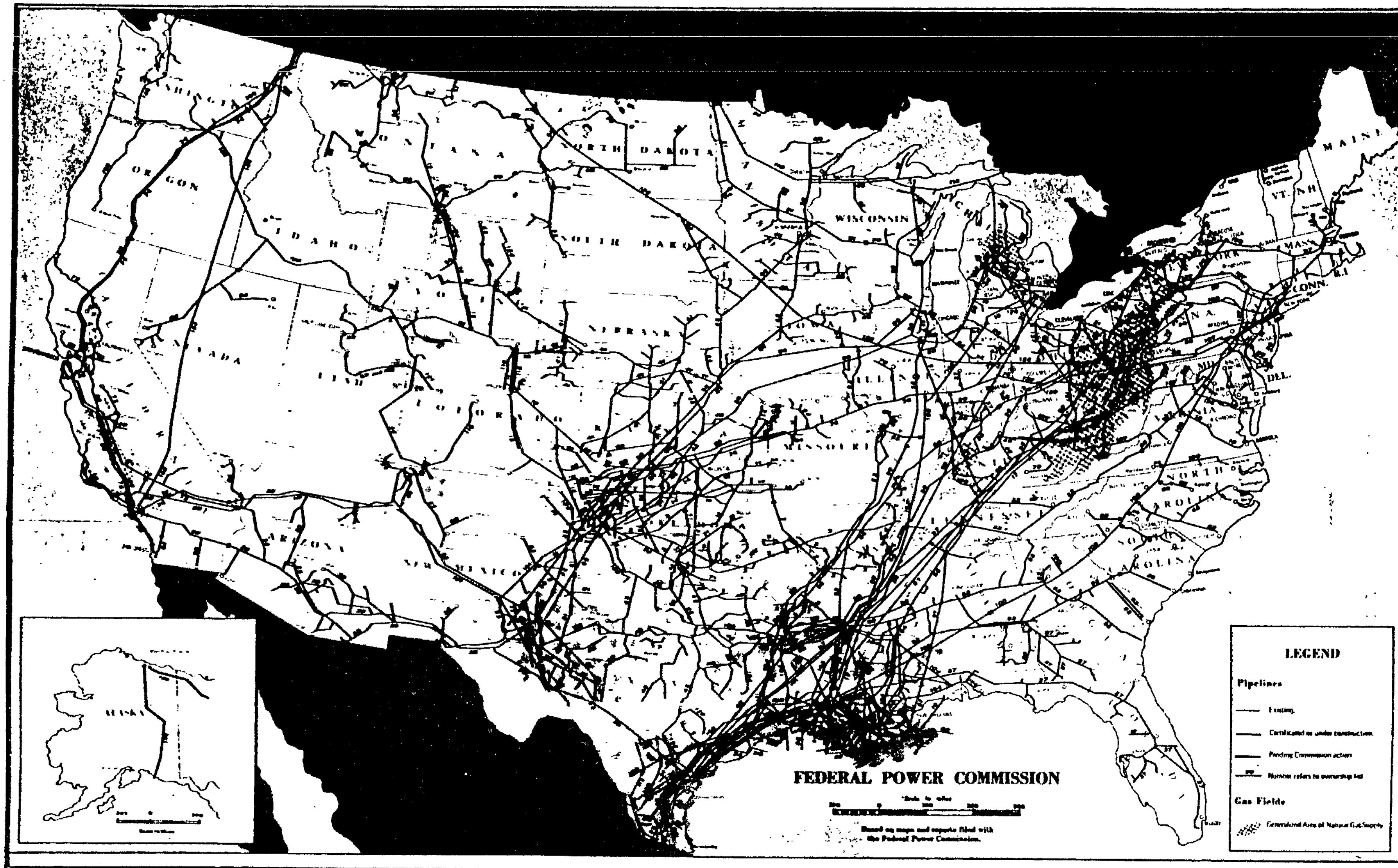


Figure 3-5(b). Major natural gas pipelines(34).

allowing the resulting high-pressure products of combustion gases to push the piston. Power is converted from linear to rotary form by means of a crankshaft.

There are two methods of igniting the fuel and air mixture. In compression ignition (CI) engines, air is compression heated in the cylinder, and diesel fuel is then injected into the hot air. Ignition is spontaneous. A different approach, however, is used for gasoline or gas-fueled engines where combustion is initiated by the spark of an electrical discharge. Therefore, these units are spark ignition (SI) engines. Usually the fuel is mixed with the air in a carburetor (for gasoline) or at the intake valve (for natural gas), but occasionally the fuel is injected into the compressed air in the cylinder. Although all diesel-fueled engines are compression ignited and all gasoline and gas-fueled engines are spark ignited, gas can be used in a compression ignition engine if a small amount of diesel fuel is injected into the compressed gas-air mixture to initiate burning. Such "dual fuel" engines are usually designed to burn any mixture ratio of gas and diesel oil, from 6- to 100-percent oil (based on heating value).

CI engines can operate at a higher compression ratio (ratio of cylinder volume when the piston is at the bottom of its stroke to volume when it is at the top) than SI engines because fuel is not present during compression and hence there is no danger of premature auto-ignition. Since the thermal efficiency of an engine increases with increasing pressure ratio (and pressure ratio varies directly with compression ratio), CI engines are more efficient than SI ones. This increased efficiency is gained at the expense of poorer acceleration and a heavier structure to withstand the higher pressures.

In addition, engines may be described by the number of strokes per cycle (two or four) and the method of introducing air and fuel into the cylinder.

In the four-stroke cycle, the sequence of events may be summarized as follows (see Figure 3-6)(37):

1. Intake stroke -- suction of the air or air and fuel mixture into the cylinder by the downward motion of the piston
2. Compression stroke -- compression of the air or air and fuel mixture, thereby raising its temperature
3. Ignition and power (expansion) stroke -- combustion and consequent downward movement of the piston with energy transfer to the crankshaft
4. Exhaust stroke -- expulsion of the exhaust gases from the cylinder by the upward movement of the piston

This description applies to a naturally aspirated engine, which utilizes the vacuum created behind the moving piston to suck in the fresh air charge. However, many engines now pressurize the air cylinder. This may be done with either a turbocharger or a supercharger. The turbocharger is powered by a turbine that is driven by the energy in the relatively hot exhaust gases, while a supercharger is driven off the engine crankshaft. Air pressurization is used to increase the power density, or power output per unit weight (or volume) of the engine. Since the density of air increases with pressure, the mass of air that can be introduced into the cylinder increases with pressure. Furthermore, since the air-to-fuel ratio at maximum power is fixed by combustion requirements, more fuel can be introduced into the cylinder with high pressure air than with atmospheric pressure air; hence, more power can be obtained from a given cylinder configuration. As the air pressure is

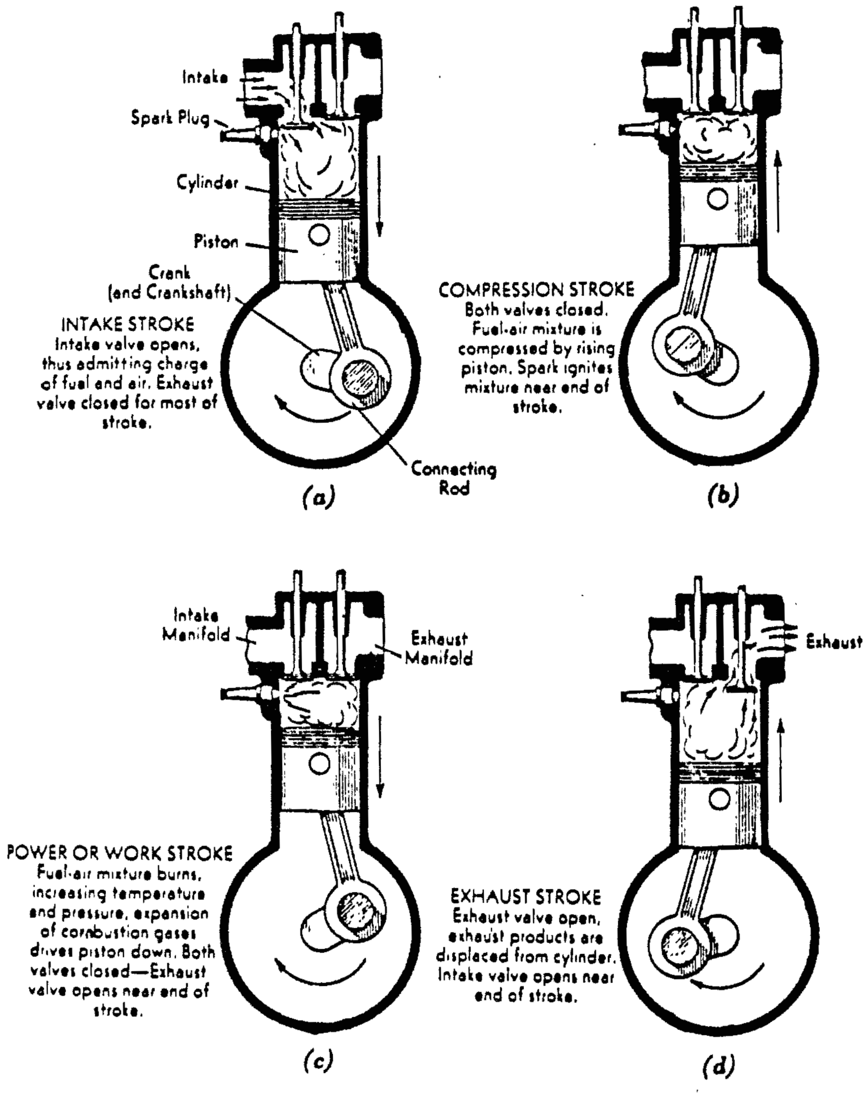


Figure 3-6. The four-stroke, spark-ignition (SI) cycle. Four strokes of 180° of crankshaft rotation each, or 720° of crankshaft rotation per cycle(37).

increased, its temperature is also raised. Therefore, the pressurized air is often cooled before entering the cylinder to further increase power. This process is called intercooling or aftercooling. In fact, all high power turbocharged natural gas-fueled engines are intercooled to prevent premature auto-ignition of the fuel and air mixture(38).

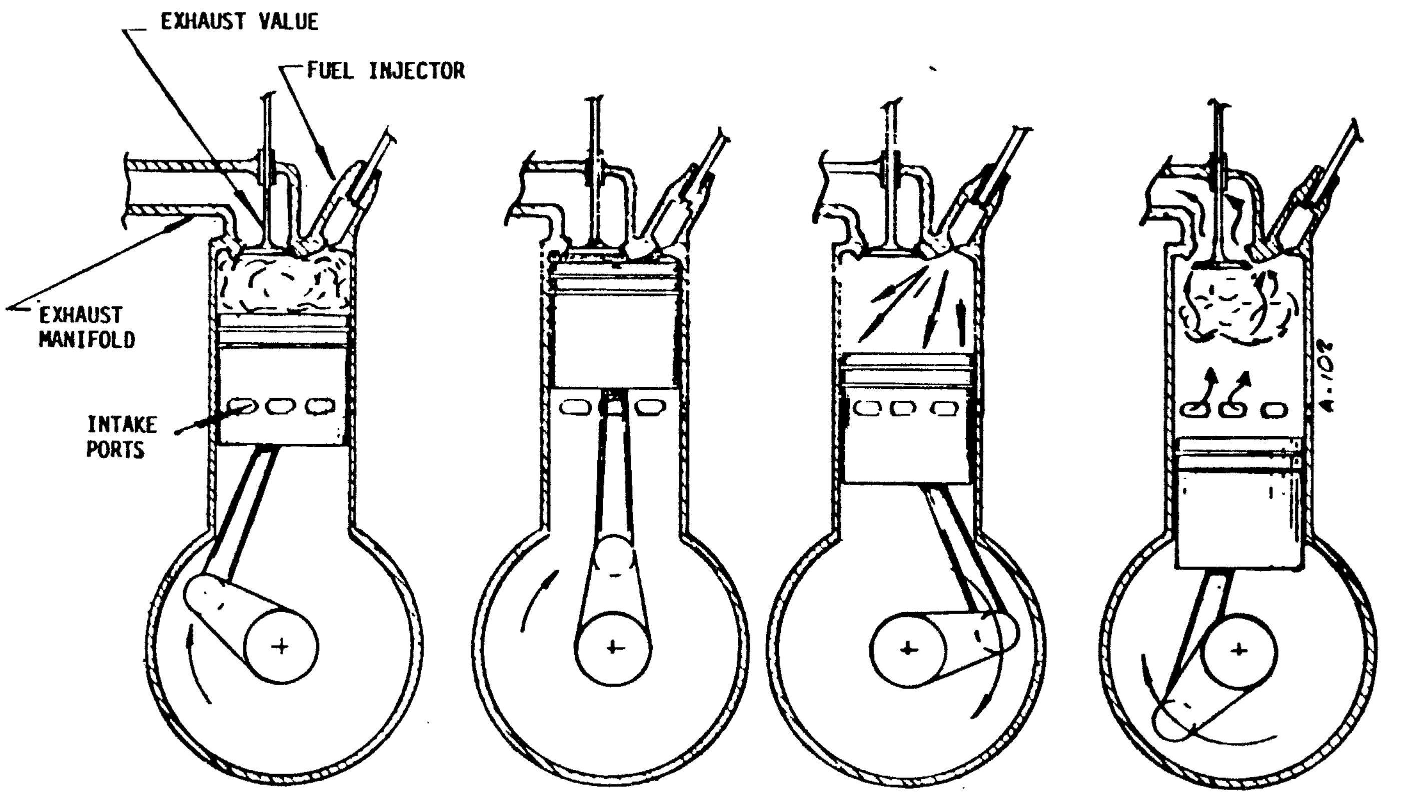
In a CI engine, fuel is injected into the cylinder near the end of the compression stroke; whereas, in an SI engine, the fuel is usually added to the air, downstream of the turbocharger if any is used, before the mixture enters the cylinder. In some SI engines (particularly large natural gas-fueled ones), the fuel is injected into the intake manifold just ahead of the valves. For CI engines or SI engines using fuel injection instead of carburetors, the fuel or fuel-air mixture can be introduced by "direct injection" into the cylinder head and communicates with the principal combustion volume. Direct injection units are also called open chamber engines because combustion takes place in the open volume between the top of the piston and the cylinder. This design contrasts with indirect injection engines, where the combustion begins in a fuel-rich (oxygen deficient) atmosphere in the smaller antechamber and then expands into the cooler, excess air region of the main chamber. These latter engines are also called divided or precombustion chamber systems (swirl chamber if the small antechamber is specially designed to promote swirl -- see Section 4.4.8). Prechamber designs can be used in carbureted engines as well, but they have not yet been applied to engines built exclusively for stationary applications.

There are three major advantages of divided chamber engines: less sensitive to fuel variations; lower stresses on the mechanical parts, such as connecting rods, crankshaft, etc., because of a lower pressure rise rate during combustion; and lower NO_x production. All three of these benefits are a result of incomplete primary combustion with an oxygen deficiency. Therefore, by comparison with open chamber designs, combustion is less sensitive to fuel droplet spray patterns, the complete combustion process takes longer, and less oxygen is available to combine with nitrogen in the initial high-temperature reaction. These benefits are attained at the expense of up to 30 percent greater heat rejection (due to the surface area added by the antechamber) and consequent 5- to 8-percent fuel penalty[39,40].

In a two-stroke design, the power cycle is completed in one revolution of the crankshaft as compared to two revolutions for the four-stroke cycle (see Figure 3-7). As the piston moves to the top of the cylinder, air, or an air and fuel mixture, is compressed for ignition. Following ignition and combustion the piston delivers power as it moves down. Eventually it uncovers the exhaust ports (or exhaust valves open). As the piston begins the next cycle, exhaust gas continues to be purged from the cylinder, partially by the upward motion of the piston and partially by the scavenging action of the incoming fresh air. Finally, all ports are covered (and/or valves closed), and the fresh charge of air or air and fuel is compressed.

Air charging in two-stroke designs is often accomplished by means of a low-pressure blower, which also aids in purging the exhaust gases; such systems are called blower-scavenged. Naturally aspirated and turbocharged (or supercharged) systems are also common.

The main advantage of two-stroke engines is their horsepower-to-weight ratio as compared to four-stroke prime movers when both operate at the same speed. This is, of course, due to the fact that the two-stroke design has twice as many power strokes per unit time as the four-stroke. In addition, if ports are used instead of valves, the mechanical design of the engine is



COMPRESSION

A. INTAKE PORTS COVERED

B. EXHAUST VALVE CLOSES

COMBUSTION

A. FUEL ENTERS CYLINDER BY INJECTION

B. COMBUSTION BY AUTO-IGNITION

POWER STROKE

A. PISTON MOVES DOWN

B. POWER DELIVERED TO CRANKSHAFT

SCAVENGING

A. AIR BLOWN INTO CYLINDER

B. EXHAUST GASES PURGED

Figure 3-7. Cylinder events for a two-stroke blower-scavenged IC engine.

simplified. However, combustion cannot be controlled as well, and excess air is needed to purge the cylinder. Therefore, these engines tend to be slightly less efficient, and uncontrolled models tend to emit slightly more pollutants (primarily unburned hydrocarbons), than do their four-stroke counterparts(41).

3.2.2 Emission Sources and Types in IC Engines

Most of the pollutants from IC engines are emitted through the exhaust. However, some hydrocarbons escape from the crankcase as a result of blowby (gases which are vented from the oil pan after they have escaped from the cylinder past the piston rings) and from the fuel tank and carburetor because of evaporation.[3/] Nearly all of the hydrocarbons from diesel (compression ignition) engines enter the atmosphere from the exhaust(43). Crankcase blowby is minor (0.18 g/hp-hr HC + NO_x, 0.0063 g/hp-hr CO) because hydrocarbons are not present during compression of the charge(44,45). Evaporative losses are insignificant in diesel engines due to the low volatility of diesel fuels(46). In general, evaporative losses are also negligible in engines using gaseous fuels because these engines receive their fuel continously from a pipe rather than via a fuel storage tank and fuel pump(47). In gasoline-fueled engines, however, 20 to 25 percent of the total hydrocarbon emissions from uncontrolled engines come from crankcase blowby and another 10 to 15 percent from evaporation of the fuel in the storage tank and the carburetor (divided approximately equally between the two)(48).

[3/] Until recently it was thought that HC evaporative controls on automobile fuel tanks and carburetors reduced emissions from these sources to about 5 percent of the uncontrolled levels. However, further, more comprehensive testing has shown that controlled vehicles actually allow HC to be evaporated at approximately 70 percent of the rate from uncontrolled cars (42).

However, crankcase blowby emissions can be virtually eliminated through the simple expedient of the positive crankcase ventilation (PCV) valve.

The primary pollutants from internal combustion engines are oxides of nitrogen (NO_x), hydrocarbons and other organic compounds (HC), carbon monoxide (CO), and particulates, which include both visible (smoke) and nonvisible emissions. NO_x formation is directly related to high pressures and temperatures during the combustion process and to the nitrogen content of the fuel. The other pollutants, HC, CO, and smoke, are primarily the result of incomplete combustion. Specific descriptions of these formation mechanisms in SI and CI engines are discussed in Chapter 4. Ash and metallic additives in the fuel also contribute to the particulate content of the exhaust, but usually do not cause the exhaust to be visible. Although the odor of the exhaust (primarily a problem with diesels) is believed to be the result of some organic compounds in this exhaust (particularly aldehydes), a precise relation has not yet been established between exhaust and odor(49-52). Considerable research has led to the belief that aldehydes may be one cause of odor in diesel and gasoline exhaust. Aromatic hydrocarbons and other unburned elements of fuel, as well as sulfur species, may also be a source of odor(53).

Oxides of sulfur, lead, and other metals also appear in the exhaust from IC engines. The sulfur compounds, mainly SO_2, are directly related to the sulfur content of the fuel. Most of the metals are fuel additives and generally leave the engine as particulates. Lead is commonly added to gasoline lines as an antiknock agent. However, laed-free gasoline is now available nationwide, and leaded gasoline may not be marketed in quantity in the future. Although various metals, particularly barium, are sometimes added to diesel fuel as a smoke suppressant, their use is not recommended by most engine manufacturers because of possible adverse effects on the engines and on the environment[54,55].

3.2.2.1 Emission Characteristics -- Spark Ignition Engines

Typical uncontrolled emission rates (23-mode composite)[56] for four-stroke naturally aspirated gasoline engines in g/brake hp-hr are: 2.4 to 13.1 for HC; 30.4 to 89.8 for CO; and 7.9 to 13.9 for NO_x[57]. Similarly, brake specific emission rates from all types of spark-ignited gas engines (two- and four-stroke, naturally aspirated and turbocharged engines) are: 0.4 to 1.9 for HC; 0.2 to 0.6 for CO; and 15.0 to 32.0 for NO_x(58).

Hydrocarbon emissions from these engines may be partially burned or completely unburned. Three general mechanisms are believed to be the cause of these pollutants[59]. The first, wall quenching, arises when fuel droplets impinge on the cylinder walls and are cooled below the ignition temperature. Whether the hydrocarbons remain unburned or are partially oxidized depends upon the wall temperature, the droplet temperature at impingement, and the combustion intensity nearby. The second source of HC emission, incomplete combustion, can be caused by one, or more, of the following:

a. Change in charge homogeneity
b. Incorrect air-to-fuel ratio (too rich or too lean)
c. Low temperature of the incoming air-fuel mixture
d. Failure of the engine to purge all the products of combustion
e. Defective ignition system

Scavenging, or the displacement of exhaust gases in the cylinder by a fresh charge of air or air and fuel, is the third mechanism resulting in HC emissions. It is especially a factor in carbureted two-cycle engines, where part of the intake charge is swept out with the exhaust completely escaping the combustion process.

Carbon monoxide emissions are also related to incomplete combustion. CO is formed during combustion and then normally combines with oxygen atoms (dissociated oxygen molecules) to form CO_2, the final product. This last reaction can only proceed at a high temperature (above 900°K)[60]. Therefore, if the overall mixture is rich or if sufficient oxygen is not available during and after combustion, before the products of combustion cool below 900°K, CO will be formed. Thus, both CO and HC are due to improper air-fuel mixing and improper introduction into the chamber. Their generation is also a measure of the inefficiency of the combustion process.

NO_x formation, on the other hand, increases with increasing combustion efficiency. Its generation is governed by the pressure, temperature and oxygen concentration during combustion. Nitrous oxide (NO) is generated in the cylinder when both the N_2 and the O_2 molecules dissociate into free atoms at the high temperature encountered during combustion, and then recombine to form NO. The reaction rate toward NO increases exponentially with temperature. Maximum temperatures occur when the air-to-fuel ratio is just above stoichiometric. NO_2 and other NO_x compounds form later in the presence of additional oxygen. NO_x emissions are particularly high from gas engines, which burn fuel with an excess of oxygen. In any engine, as the air-to-fuel ratio decreases from stoichiometric, NO_x formation decreases because of a lack of excess oxygen. As the air-to-fuel ratio increases from stoichiometric, NO_x formation first increases due to the presence of more oxygen. Further increases in air flowrate cause a decline in peak combustion temperature and a consequent decrease in NO_x formation (Figure 3-8)[61].

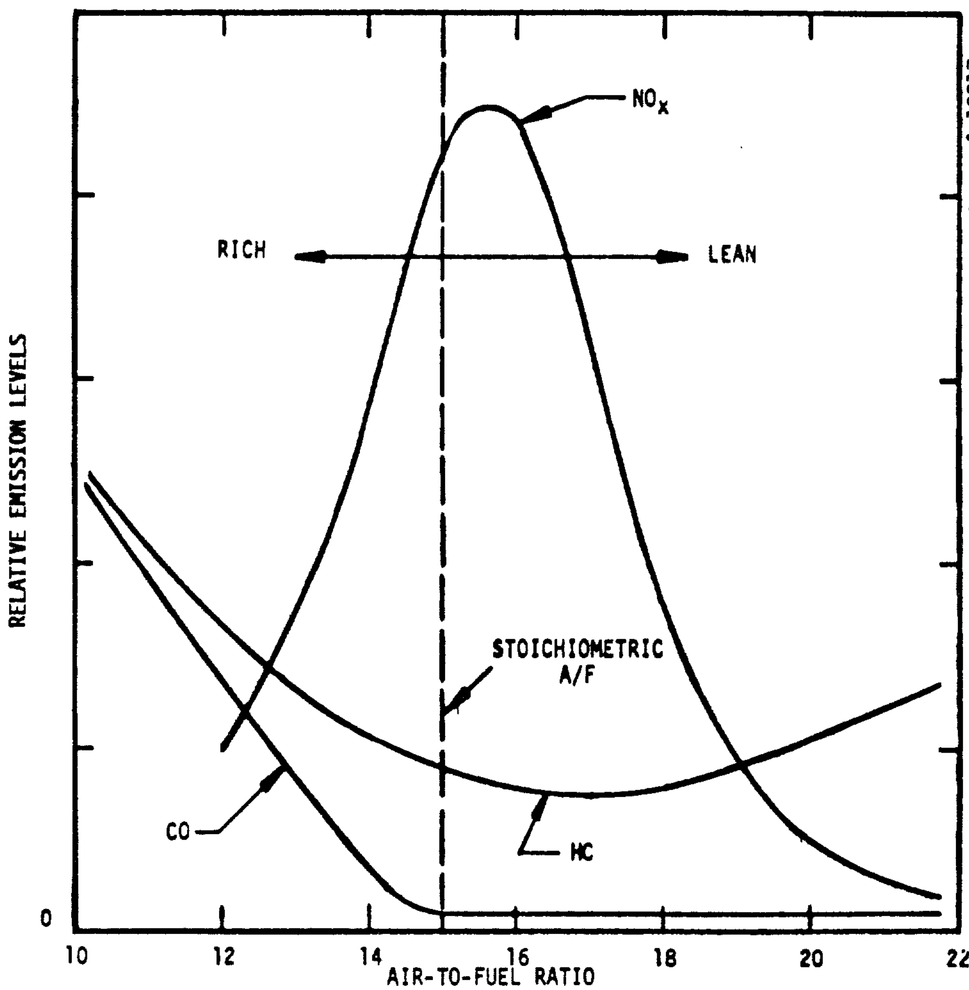


Figure 3-8. Effect of A/F ratio on emissions from a gasoline engine[61].

Visible emissions (smoke) are generally insignificant in gasoline and gas engines. Particulates, however, may be present, especially as a result of rich fuel mixtures or impurities in the fuel. On a mass basis, the majority of particulate emissions from gasoline engine exhausts are now the result of lead in the fuel(62). When smoke is present, particulate levels are high, but the converse does not necessarily hold; significant quantities of non-light-scattering particulates can be emitted in the absence of visible emissions. The bluish smoke that is sometimes observed, particularly in older, poorly maintained engines, is caused by incomplete combustion of crankcase or lubricating oil which is forced past worn piston rings into the cylinder.

3.2.2.2 Emission Characteristics -- Compression Ignition Engines

As mentioned previously, essentially all the pollutants from diesel or dual-fuel engines are emitted through the exhaust (i.e., evaporative and crankcase ventilation emissions are negligible). Again HC, CO, NO_x and particulate matter are the pollutants of major concern. The formation of these pollutants is complex and not completely understood. Typical emission rates (specific) for all types of uncontrolled diesel engines (including four-stroke, either naturally aspirated or turbocharged and open- or divided-chamber, and two-stroke, either blower-scavenged or turbocharged) in g/brake hp-hr are: 0.1 to 2.9 for HC; 0.3 to 14.6 for CO; and 2.1 to 17.1 for NO_x(63). Unburned and partially burned hydrocarbons are formed in CI engines by many of the same mechanisms that contribute to their generation in SI engines. There are, however, two major differences: the fuel injection system and the combustion chamber design. Fuel is introduced into the cylinder of a CI engine by a fuel injector, independently of the air, whereas a fuel and air mixture generally is allowed to flow into the cylinder of an SI engine through the intake valve or port. Therefore, the fuel-air mixture in a CI engine is heterogeneous by design. However, the fuel distribution within the cylinder can be controlled somewhat by the design of the injector, and therefore, both wall quenching and large-scale inhomogeneities can be minimized(64,65). Hence, both the fuel injection system and the combustion chamber design are major factors which determine the quantity of HC emitted(66,67).

Since the carbon monoxide results from incomplete combustion in the absence of sufficient oxygen, and since CI engines are designed to operate lean (excess air), CO emissions are usually low.

Conditions within the cylinder of a CI engine during combustion favor the production of NO_x, because it is formed at high temperatures and pressures in the presence of oxygen. However, high temperatures and pressures lead to high thermal efficiency (low fuel consumption) and high air-to-fuel ratios lead to low smoke, HC, and CO generation(68,69). Therefore, design compromises are usually required when an attempt is made to reduce or limit NO_x emissions.

Smoke occurs in three forms: white (cold) smoke, blue, and black (hot smoke)(70). The white smoke consists primarily of unburned liquid fuel or lubricating oil and occurs at low load or idle conditions. Blue smoke is caused by lubrication oil that leaks into the combustion chamber. Black smoke consists of carbon particles that result from incomplete combustion at high temperatures. Incomplete combustion is possible under high loads (low air-to-fuel ratio) and poor mixing of the fuel-air mixture. Emission rates of white and blue smoke are usually not measured by manufacturers. They generally report black smoke emission rates of less than 10-percent opacity

(1.18 milligrams per cubic foot for a 16-inch diameter stack) for a well maintained engine(71-73).

3.2.2.3 Effects of Variables on Emissions

Many variables affect emissions from IC engines. Table 3-3(74) lists these variables, divided into three categories. Those listed as "Design" are features that are normally determined by the manufacturer. The "Operation Adjustment" category represents variables that can be altered by the user. Since these variables can be adjusted to reduce emissions, they are discussed in more detail in Chapter 4, Emission Control Techniques. The last column lists those variations in ambient air conditions which affect emissions.

3.2.2.3.1 Size

Engine size has a definite effect on emissions. Large-bore, low- to medium-speed engines, independent of the fuel type (diesel, dual fuel, natural gas), have certain common emission characteristics. These large-bore engines are designed for low fuel consumption (i.e., high thermal efficiency). Furthermore, their low rotational speed maintains the products of combustion near peak temperature for relatively long periods of time. These slow speeds allow the designer to "tailor" his fuel injection schedule for optimum results, and the large combustion volumes also give him more aerodynamic design freedom than is available to the manufacturer of small- or medium-bore engines(75). Furthermore, these large engines are designed to operate under steady conditions and almost always at more than 50 percent of rated power; therefore, they do not have to contend with acceleration/deceleration or low power requirements(76,77).

TABLE 3-3. FACTORS THAT AFFECT EMISSIONS FROM RECIPROCATING ENGINES[74]

Design	Operation Adjustment	Ambient Conditions
Surface to volume ratio	Air-to-fuel ratio	Pressure (altitude)
Bore and stroke	Torque (mean effective pressure)	Temperature
Valve overlap	Speed	Humidity
Displacement/cylinder	Spark timing	
Strokes/power cycle	Fuel injection timing	
Chamber design	Fuel properties	
Compression ratio	Lubrication and maintenance	
Air charging:		
Naturally aspirated		
Blower scavenged		
Turbocharged		
Fuel charging:		
Direct injection		
Indirect injection		
Carbureted		
Engine cooling		

3.2.2.3.2 Load and Air-to-Fuel Ratio

Load and air-to-fuel (A/F) ratio variations have a significant effect on NO_x, CO, HC, and particulate emissions. In general, diesel compression ignition engines exhibit decreasing brake specific emissions of NO_x with increasing load (i.e., the lower A/F ratio and higher power output overcome the potential for increased NO_x production due to higher temperatures)(78). CO emissions first decrease with increasing load (equivalent to increasing temperature) and then increase dramatically as maximum load is approached. Incomplete combustion with the decreasing A/F ratio (increasing load) is largely responsible for the final trend reversal. Brake specific HC emissions, likewise, decrease with increasing load as a result of increasing temperature(79). Small (less than 15 hp) SI engines (gasoline) exhibit relatively high HC and CO emissions, particularly at low load operation(80).

Figure 3-8 illustrated the effect of A/F ratio upon emissions from spark ignition engines. NO_x and CO emissions are very definitely related to A/F ratios. These trends hold throughout the load range for an SI engine. Note that an engine whose A/F ratio is adjusted to yield lowest HC and CO emissions has a maximum NO_x emission. An uncontrolled engine that is adjusted to near-stoichiometric conditions will emit relatively large quantities of NO_x, whereas one that is run either lean or rich will produce significant quantities of HC and CO (assuming other modifications or control technologies are not applied to control HC and CO).

3.2.2.3.3 Climate and Location

All the effects of variations in climate and geographic location on emissions are reflected by changes in the pressure, temperature, and humidity of the ambient air. Pressure and temperature variations alter the density of the inlet air charge and, hence, change the air-to-fuel ratio. (The consequences of changing A/F ratio have just been discussed.) Secondarily, they change the peak combustion pressure and temperature in proportion to their variations in the ambient air. Since NO_x production is a strong function of temperature, variations of the inlet air temperature have a noticeable effect on NO_x emissions. In general, humidity has little effect upon HC and CO emissions(81,82). NO_x emissions, however, are significantly affected by humidity variations because the water absorbs some of the heat generated during combustion, thereby reducing the peak combustion temperature. Although the qualitative effects of varying ambient conditions of emissions are known, a reliable quantitative relationship exists only for the effects of humidity on NO_x production from vehicular gasoline and diesel engines(83,84). Since these relationships are based on experiments, and cannot be predicted analytically, they are applicable only to stationary engines which are similar to vehicular models. Similar relationships have not yet been developed for high-power, low- and medium-speed engines. Since the combustion characteristics of these large engines are different than those of the vehicular size units, there is no a priori justification for applying the above-mentioned relationship to the larger engines.

3.2.2.3.4 Fuel Effects

Significant variations in the emissions of reactive HC may be due to the fuels used. As mentioned previously, HC emissions from SI gasoline engines originate from the fuel system through evaporation, from the exhaust due to combustion, and from crankcase blowby. In general, fuel evaporation is proportional to fuel volatility(85). Effective systems are now used to control evaporative emissions and crankcase blowby on automobiles. Studies have shown that exhaust HC emissions from unleaded fuels have a higher photochemical reactivity than do those from base leaded fuels, assuming no catalytic converters are used(86).

Diesel fuels used in CI engines may display some variation in density and viscosity which, in turn, affects the A/F ratio and fuel spray pattern in the cylinder. If only a single fuel is used, the engine is "tuned" for it, but is is unlikely that engines would be "retuned" for each fuel if fuel types were frequently changed.

Particulate and smoke generation tends to increase with increasing Cetane number. The relationship between smoke and fuel volatility is unclear because it is difficult to separate the effects due to volatility changes from those due to Cetane number(87).

3.2.3 State and Local Regulations

Most states and localities restrict visible emissions from stationary sources, including reciprocating internal combustion engines, to less than 20-percent opacity and many are now requiring that plumes from new installations have less than 10-percent capacity(88). These limits usually do not present a problem to the owner who operates and maintains his engine correctly. However, since many NO_x control systems tend to increase smoke, engines may not be able to meet these opacity limits if very stringent NO_x standards are proposed.

The only known restrictions on gaseous emissions from reciprocating IC engines (i.e., excluding gas turbines, which have been specifically regulated by some localities) are in the Los Angeles area. Several control districts in this basin limit NO_x emissions (expressed as NO_2) to 140 lb/hr (63.5 kg/hr)(89-91). Typical uncontrolled engines emit 10 to 20 g/hp-hr (see

Chapter 5). At 15 g/hp-hr these NO_x limits would prevent a user from buying an uncontrolled engine that is larger than about 4250 hp (3200 kW). As is also shown in Chapter 5, emissions from a few large-bore engines can be brought down to levels as low as 8 g/hp-hr by control techniques that can be applied in the near term. Such engines could be as large as 7950 hp (6000 kW) and still be allowed to operate in the Los Angeles basin.

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(91) "Rules and Regulations," County of Los Angeles Air Pollution Control District, Los Angeles, California, Rule 67.

CHAPTER 4

EMISSION CONTROL TECHNIQUES

Stationary reciprocating IC engines emit primarily NO_x, CO, HC, and particulates (see Section 3.2.2). All of these emissions are criteria pollutants; that is, National Ambient Air Quality Standards (or maximum allowable ambient concentrations) have been established for these pollutants under Sections 108 and 109 of the Clean Air Act. As indicated in Section 9.1, EPA has placed a high priority on the control of NO_x emissions from stationary sources since these emissions are projected to increase about 25 percent by 1985 despite the application of best available control technology to new sources. Moreover, NO_x is the most significant pollutant emitted from stationary IC engines, accounting for more than 16 percent of the total NO_x emitted from stationary sources. In addition, as will be shown in Section 9.1, approximately 75 percent of the NO_x emissions from installed stationary reciprocating IC engines (1974) was contributed by units larger than 500 hp. This chapter, therefore, emphasizes the control of NO_x, and control techniques are discussed which have been applied, or can potentially be applied, to large, stationary IC engines.

This chapter is divided into six sections. The formation of the pollutants emitted by IC engines is briefly discussed in Section 4.1 to aid in understanding emissions control. Section 4.2 discusses the effect of ambient conditions (humidity, temperature, and pressure), and measurement practices upon reported emission levels. Specifically, Section 4.2.1 describes how ambient correction factors will be used to adjust reported NO_x data to standard conditions, and Section 4.2.2 discusses and compares the measurement practices of the manufacturers who reported emission data. The uncertainty of each stationary IC engine manufacturer's measurement practice is estimated relative to EPA's proposed approach for 1979 and later model heavy-duty diesel and gasoline engines. These uncertainties will be used to place boundaries on estimates of average emissions from engines of the same fuel type, but different manufacturer.

Section 4.3 presents uncontrolled (baseline) emission levels for large stationary IC engines. In addition to these data, sales-weighted averages of uncontrolled NO_x levels (by horsepower, excluding standby applications) will be presented.

Section 4.4 discusses specific control systems to reduce NO_x emissions. The discussion considers how each control system works: its effectiveness, resulting fuel penalties, effects on other emissions (HC, CO, smoke), technical limitations to its application, and cost implications (fuel, hardware, and maintenance). The emission data are corrected to standard ambient conditions where possible (see Section 4.2.1).

Finally, control systems for HC and CO emissions are discussed in Section 4.5, and techniques to reduce visible emissions are reviewed in Section 4.6. As stated in Section 4.5, there has been little effort by manufacturers of large-bore engines to reduce emission rates of HC and CO from their engines, primarily because there are currently no regulations affecting these emissions and they are already quite low. Moreover, the application of NO_x control systems has, in general, only a small effect on HC and CO levels (see Section 4.4.12). Furthermore, several manufacturers have achieved reductions in HC and CO levels as a consequence of their efforts to reduce visible emissions and improve the fuel economy of their diesel engines. These practices are discussed in both Sections 4.5 and 4.6.

4.1 POLLUTANT FORMATION

The combustion of fuel and air in the cylinder of a reciprocating engine results in the formation of a number of chemical compounds which can "cause or contribute to the endangerment of public health or welfare" when emitted to the atmosphere. The ones of most concern fall into the general categories of nitrogen oxides (NO_x), carbon monoxide (CO), sulfur oxides (SO_x), various hydrocarbons and organic compounds (HC), and particulates. Additionally, smoke and odorous fumes may be produced, chiefly in compression ignition engines. The following paragraphs discuss the formation of these pollutants in stationary reciprocating IC engines.

4.1.1 Nitrogen Oxides (NO_x)

Virtually all NO_x emissions originate as nitric oxide (NO) resulting from oxidation of nitrogen during the combustion process. Both N_2 and O_2 dissociate into atomic N and O, respectively, at the pressures and temperatures encountered in the cylinder. These atoms combine to form NO, part of which is then further oxidized in the engine and exhaust system to form NO_2. The remaining NO is exhausted into the atmosphere, where, in the presence of additional oxygen, it also reacts rapidly to form the more stable NO_2. The NO_2 molecules, however, can be dissociated by ultraviolet light from the sun. Additional reactions involving the ultraviolet light from the sun, nitrogen and hydrocarbon effluents, and atmospheric oxygen produce polynuclear aromatics and other components of photochemical smog.

Most of the nitrogen enters the engine as molecular N_2 in the air, since natural gas and the premium distillate fuels generally used in reciprocating engines contain little fuel-bound nitrogen. Oxides of nitrogen that come from N_2 in the air are called thermal NO_x. However, the heavier oils, such as residuals and crudes, contain significant quantities of nitrogen (e.g., up to 1 percent). When these fuels are used, as is occasionally done in large diesel engines, noticeable amounts of "fuel", or "organic", NO_x are generated. The degree of oxidation of the nitrogen in the fuel appears to be primarily a function of its nitrogen content. For gas turbines, several researchers have found that the fraction of fuel-bound nitrogen converted to NO decreases with increasing nitrogen content, although the absolute magnitude of the NO formed increases (1,2). For example, a low nitrogen fuel (0.01 percent) may have 100 percent of the fuelbound nitrogen converted to NO_x, whereas a high nitrogen fuel (1.0 percent) may have only 40 percent of it reacted to NO_x. Consequently, when residual oil is burned in turbines, more NO is produced from fuel-bound nitrogen than from N_2 present in the air(3).

Similar data are not available for reciprocating engines. However, the relative magnitudes of thermal and fuel NO_x are quite different for reciprocating engines than for turbines because the engines generate about 10 times more thermal NO_x than do the turbines. Thus, if the fuel for an engine contained 1-percent nitrogen, and all of this were converted to oxides of nitrogen, the exhaust would contain about 5 g/hp-hr NO_2. By comparison, uncontrolled large-bore diesel engines emit 7.5 to 18.7 g/hp-hr when using distillate (No. 2) containing about 0.03-percent nitrogen(4).

All data presented in this document on emissions from diesel and dual fuel engines are based on operation with No. 2 distillate (diesel oil). The possible effects of any of the controls to be discussed in this chapter on the fuel-bound nitrogen are not known. Naturally, denitrification of the fuel prior to its use will reduce the potential NO_x emissions, and this "control" is discussed briefly in conjunction with fuel desulfurization (see Section 4.4.13).

The rate of formation of thermal NO is a function of the residence time of the atomic nitrogen with atomic oxygen at elevated temperatures. The minimization of any or all of these three underlined variables forms the basis for almost all successfully demonstrated NO_x control methods. Although exhaust treatment of the NO via catalytic converters has been proposed and tested for gasoline-fueled automobiles, research into its applicability to diesel and large natural gas engines is just reaching the experimental stage(5,6). This subject is treated in more depth later (Section 4.4.9).

4.1.2 Hydrocarbons (HC)

The pollutants commonly classified as hydrocarbons are composed of a wide variety of organic compounds. They are discharged into the atmosphere when some of the fuel remains unburned or is only partially burned during the combustion process.

Most unburned hydrocarbon emissions result from fuel droplets that were transported or injected into the "quench layer" during combustion. This is the region immediately adjacent to the combustion chamber surfaces, where heat transfer outward through the cylinder walls causes the mixture temperatures to be too low to support combustion.

Partially burned hydrocarbons, on the other hand, can occur for a number of reasons(7,8):

- Poor air-fuel homogeneity due to incomplete mixing prior to, or during, combustion (i.e., local zones that are too rich or too lean). The most common cause of this is improper maintenance or design of the fuel handling system.
- Incorrect air-fuel ratios in the cylinder during combustion (i.e., the entire cylinder is too rich or too lean) due to maladjustment of the engine fuel system.
- Excessively large fuel droplets (diesel engines). This is generally the result of worn, clogged, or poorly designed injectors and/or low injection pressures.
- Low cylinder temperature due to excessive cooling through the walls or early cooling of the gases by expansion of the combustion volume caused by piston motion before combustion is completed. This condition commonly occurs in engines that have faulty temperature control systems or have excessively delayed ignition.

All of these conditions can be caused by either poor maintenance or faulty design. Therefore, the lowest emissions will be achieved only by proper maintenance of engines designed specifically for low emissions.

4.1.3 Odor

Odor in the exhaust is primarily a problem with diesel engines. Although the smell is believed to be from some of the hydrocarbons in the exhaust, a precise relation has not yet been established between exhaust and odor(9,10,11). Considerable research has led to the belief that aldehydes may be one cause of odor in diesel (and gasoline) exhaust. Aromatic hydrocarbons and other unburned elements of fuel, as well as sulfur species, may also be a source of odor(12).

Subjective evaluation of automobile diesel odors has shown that installing sac injectors (see Section 4.5) and catalytic mufflers for hydrocarbon control also reduces odor levels(13).

Standards for odor are not being developed at this time because odor levels are low in the vicinity of large-bore engines (in part due to their relatively tall stacks), no control technology has been demonstrated specifically for odors, and no readily implemented method exists for measuring odor levels.

4.1.4 Carbon Monoxide (CO)

Carbon monoxide is an intermediate combustion product that appears in the exhaust when the reaction of CO to CO_2 cannot proceed to completion(14). This situation occurs if there is a lack of available oxygen, if the gas temperature is too low, or if the residence time in the cylinder is too short(15). The oxidation rate of CO is limited by reaction kinetics and, as a consequence, can be accelerated only to a certain extent by improvements in air-fuel mixing during the combustion process(16).

4.1.5 Smoke and Particulate Matter

White, blue, and black smoke may be emitted from IC engines. Liquid particulates appear as white smoke in the exhaust during an engine cold start, idling, or low load operation. These are formed in the quench layer adjacent to the cylinder walls, where the temperatures are not high enough to ignite the fuel. They consist primarily of raw fuel with some partially burned hydrocarbons and lubricating oil(17). White smoke emissions are generally associated with older gasoline engines and are rarely seen in the exhaust from diesel or gas-fueled units. They cease when the engine reaches its normal operating temperature and can be minimized during low demand situations by proper idle adjustment.

Blue smoke is emitted when lubricating oil leaks, often past worn piston rings, into the combustion chamber and is partially burned(18). Proper maintenance is the most effective method of preventing these emissions from all types of IC engines.

The exact formation mechanisms of black smoke are still not fully understood, although a number of models and theories have been offered to explain its formation in diesel engines(19,20). It is generally agreed, however, that the primary constituent of black smoke is agglomerated carbon particles (soot). These form in a two-step process in regions of the combusting mixture that are oxygen deficient(21). First the hydrocarbons decompose into acetylene and hydrogen in the high temperature regions of the cylinder. Then, when the local gas temperature drops as the piston moves down and the gases expand, the acetylene condenses and releases its hydrogen atoms. As a result, pure carbon particles are created. This mechanism of formation is associated with the low air-to-fuel ratio conditions that commonly exist at the core of the injected fuel spray, in the center of large individual fuel droplets, and in fuel layers along the walls(22). The formation of particules from this source can be reduced by designing the fuel injector to provide for an even distribution of fine fuel droplets such that they do not impinge on the cylinder walls.

Once formed, the carbon will combine with oxygen to form CO and CO_2 if it is still at an elevated temperature. Since the temperature of the exhaust system is too low for this oxidation to occur, soot that leaves the combustion chamber before it has had the opportunity to oxidize completely will be discharged as visible particles.

Becauase soot formation is very sensitive to the need for oxygen, its discharge is greatest when the engine is operating at rich fuel/air ratios, such as at rated power and speed. Therefore, naturally aspirated engines are likely to have higher smoke levels than turbocharged engines, which operate at leaner fuel/air ratios.

4.1.6 Sulfur Dioxide (SO_2)

Sulfur dioxide emissions are a function of only the sulfur content in the fuel rather than of any combustion variables. In fact, during the combustion process essentially all the sulfur in the fuel is oxidized to SO_2.1/ Manufacturers of diesel engines currently recommended that users burn only fuels which contain less than 0.5 percent sulfur(23).2/ This practice is suggested to minimize corrosion, but it also results in SO_2 emission levels of less than 2 g/hp-hr from most internal combustion engines.3/ Nevertherless, fuel sulfur content may range as high as 1.0 percent according to ASTM fuel specifications for No. 2 or No. 2D distillate oils(26). These distillates are the fuels most commonly burned in stationary, large-bore diesel engines. The refining of crude oil into the No. 1 and No. 2 distillate,

1/Presumably the ratio of SO_3 to SO_2 in the exhaust of IC engines is a function of the combustion process, as it is in boilers. Data are not available to clarify this question, and in any case, the total sulfur emitted is equal to the sulfur contained in the fuel.

2/This level, which corresponds to approximately 0.7 lbm/MBtu based on heat input, is nearly one-half the value specified in the standards of performance for new oil-fired steam generators (1.2 lbm/MBtu, assuming 40 percent efficiency)(24). Existing steam plants average 3.0 to 5.0 lbm/MBtu heat input (equivalent to 8.7 to 14.4 g/hp-hr output)(25).

3/For example, an engine firing 0.5 percent sulfur No. 2 distillate at a rate of 0.4 lb/hp-hr would emit 1.8 g/hp-hr of SO_2, assuming all the sulfur in the fuel is converted to SO_2.

however, removes most of the sulfur (which becomes concentrated in the residual oil). Thus, engine users generally have no difficulty obtaining low sulfur (less than 0.5 percent) distillates.

Similarily, natural gas that is transported by pipeline is virtually free of sulfur. That is, the gas from some sources contains very little sulfur, and supplies which do contain sulfur (usually in the form of hydrogen sulfide) must, by law, be desulfurized before the gas can be transported in interstate commerce. This chemical desulfurization technology is routinely practiced. An "emerging" industry which makes "synthetic natural gas" from oil or coal also desulfurizes it prior to sale.

However, sulfur is generally not removed from sewage digestor gas that is burned onsite by large-bore IC engines. Nevertheless, as is discussed in Section 4.4.13, engines operating on unprocessed sewage gas emit no more than 1 g/hp-hr of SO_2, which is less than the SO_2 emission rate from engines burning 0.5 percent sulfur distillates. As a result, engines fueled with distillates and gases are already relatively low in SO_x emissions. Therefore, the only potential SO_2 problem from reciprocating engines should be those units that are fueled with residual oil. As discussed in Section 4.4.13, the most feasible control technique for SO_2 reduction in an engine is the use of low sulfur fuels. Consequently, desulfurization costs are discussed in Section 8.

4.2 FACTORS THAT AFFECT REPORTED NO_x EMISSIONS LEVELS

Ambient conditions and measurement practices have been shown to affect measured NO_x emissions significantly (see Appendix C.2 and C.3). Recognizing these effects, EPA's Office of Mobile Source Air Pollution Control has required that specific ambient correction factors (to adjust measured NOx data to standard conditions of humidity and temperature) and measurement procedures be adopted by manufacturers of diesel and gasoline engines whose emissions are regulated under Title II of the Clean Air Act. For the same reasons, ambient correction factors should be used when analyzing emissions data from stationary IC engines. Likewise, consideration should be given to potential impacts on the certainty of the data due to the measurement procedures used.

To this end, Section 4.2.1 briefly reviews the effect of ambient conditions on the reported NO_x data and then recommends ambient correction factors appropriate for stationary IC engines. Section 4.2.2, in turn, discusses and compares the four different measurement practices used by the nine manufacturers of large-bore stationary IC engines. Estimates of uncertainty for three of the practices (relative to the EPA procedure) are given for the manufacturers who did not follow the EPA practice. These estimates will be used to determine the uncertainty in average emission levels that were computed from data provided by the manufacturers.

4.2.1 Effect of Ambient Humidity, Temperature, and Pressure

EPA has adopted ambient correction factors for regulated mobile sources, based on experimental studies(27,28). These factors are used to correct observed NO_x levels to values at a standard temperature (85^{o}F) and humidity (75 grains H_2O/lb dry air) for heavy duty (HD) diesel engines and to correct to standard humidity, only, for both heavy and light duty gasoline vehicles(29,30). HC and CO emissions from both CI and SI engines are also sensitive to ambient humidity, temperature, and pressure. One study of diesel engines established that variations in humidity did not affect HC, CO, or smoke levels(31). Temperature variations, however, did substantially affect

HC, CO, and smoke levels, but no corrections could be generalized. In another study of automotive spark ignition engines, ambient humidity variations were shown to affect HC levels, and temperature and pressure changes to affect HC and CO emissions(32). However, no factors could be generalized for these effects.

Although the nine large-bore engine manufacturers provided ambient conditions (barometric pressure and inlet air temperature and humidity) for much of their emission data, none of them reported emission data for a given engine operated under a systematic variation of ambient conditions. Thus, at this time it is not possible to directly derive ambient correction factors for large-bore engines. Nevertheless, the response of these large-bore engines to changes in ambient conditions is anticipated to be similar to that observed in the smaller bore diesel and gasoline engines for which ambient correction factors have been developed. In fact, one large-bore manufacturer has adopted a humidity correction factor based on smaller bore engines to correct NO_x emissions data from diesel and natural gas-fired engines (33). Although this manufacturer has not conducted a systematic study of the effect of ambient humidity on NO_x emissions, he believes the correction factor derived from data on smaller bore engines is reasonable. This belief is based on the reasonable correlation obtained by using these correction factors for a limited number of tests with his own engines.

This section will, therefore, illustrate the effect of ambient variations on reported NO_x levels, briefly review existing ambient correction factors, and recommend those suitable for application to large bore engine data. (A more detailed discussion can be found in Appendix C.2.) These ambient correction factors will then be used wherever possible in the remaining sections of this chapter to correct the data reported by the engine manufacturers.

4.2.1.1 Examples of Ambient Effects on NO_x Emissions from IC Engines

As mentioned above, several studies have shown the effect of ambient humidity on NO_x emissions(34,35,36). Figure 4-1(37) is taken from one of these studies and illustrates the variation in specific humidity each month during the year 1969 for Dearborn, Michigan. For this area, ambient humidity varied from 20 to 120 grains H_2O/lb dry air over the year. The effect of this variation on NO_x emissions is shown in Figure 4-2(38,39) using the correction factors that have been derived from experimental work for gasoline and diesel engines. The NO_x levels are shown to deviate as much as 25 percent from levels measured at standard conditions. Moreover, the corrections factors used vary significantly as well, depending on the particular study and type of emission source.

Variations in ambient temperature have also been shown to affect NO_x emissions. This effect is illustrated in Figure 4-3(40,41) and ranges from 5 to 25 percent depending on the particular study. Although the effect for diesel engines is not as large as that produced by ambient humidity variations, the change in brake specific emissions (g/hp-hr) can be significant. This is particularly true for some large-bore spark ignition engines, which emit as much as 20 g/hp-hr. For these sources a 5-percent ambient correction means a change in the reported level of about 1 g/hp-hr.

Variations in barometric pressure also can be expected to affect NO_x emission(42,43). Only one study, however, has evaluated this ambient variation(44). This investigation used carbureted gasoline engines and found NO_x variations of as much as 40 percent due to changes in ambient pressure.

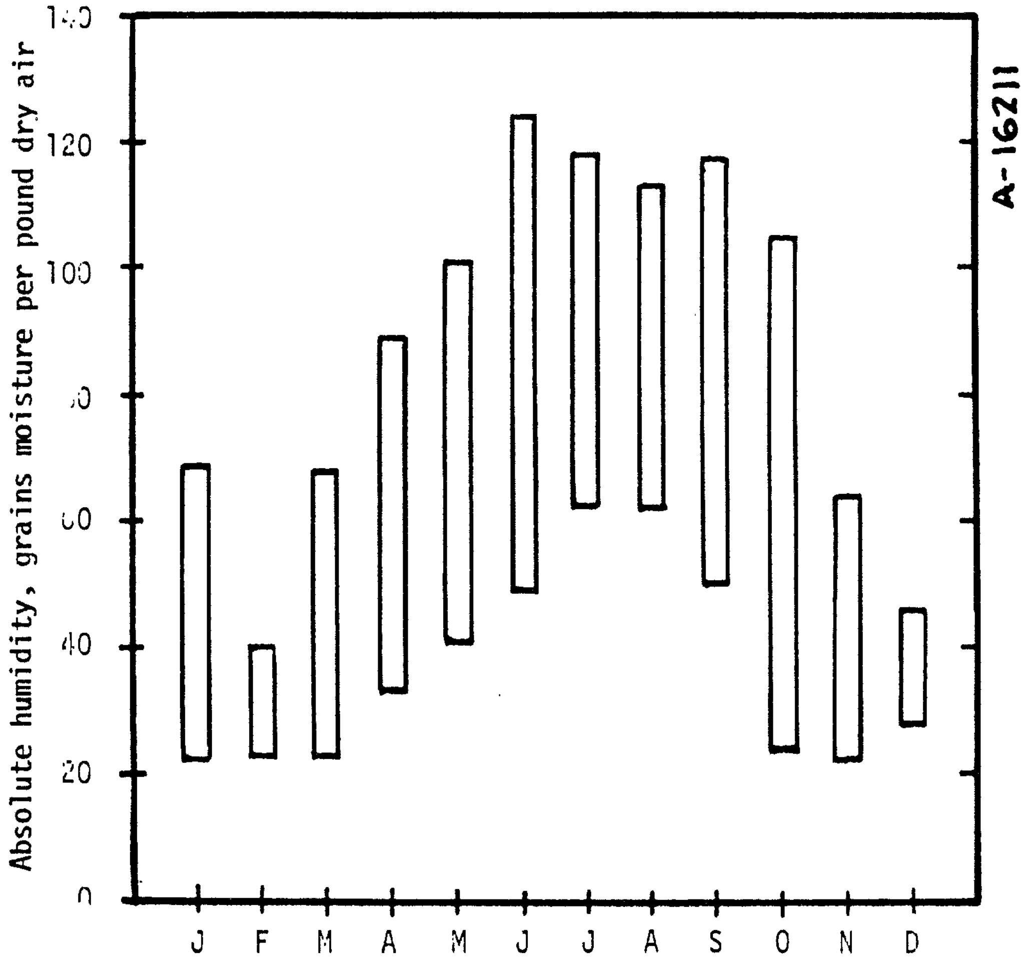
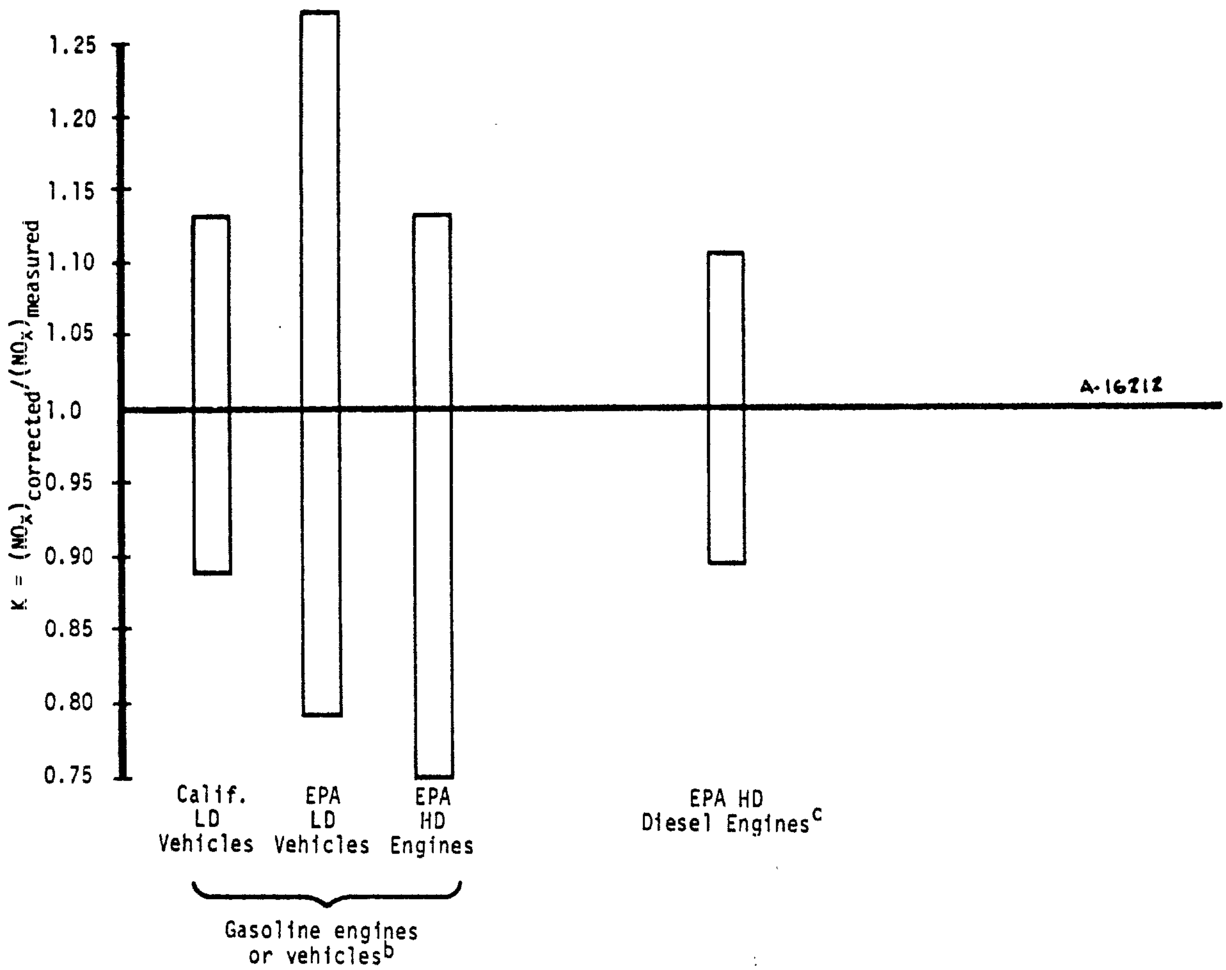


Figure 4-1. 1969 humidity range in engine test cells, Dearborn, Michigan (from Reference 37).

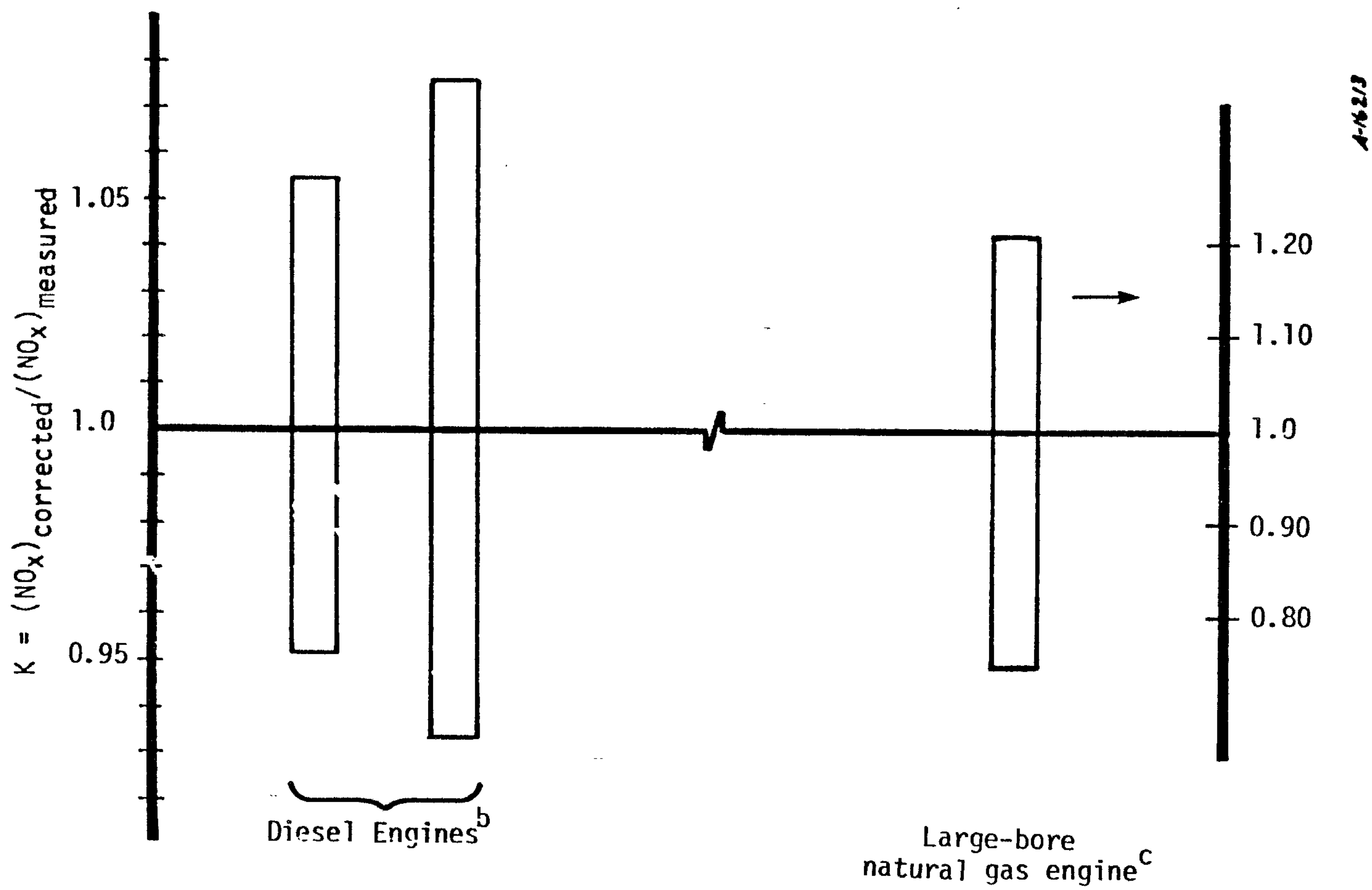


[a]Range based on humidity variation from 20 to 120 grains/lb dry air referenced to 75 grains/lb dry air standard condition. Note: K > 1.0 for humidity >75 grains/lb dry air.

[b]From Reference 38.

[c]From Reference 39.

Figure 4-2. Effect of ambient humidity on NO_x emissions for IC engines.[a]



[a] Based on range of ±30°F from standard temperature 85°F. Note: K > 1 for temperatures less than 85°F.

[b] Reference 40.

[c] Based on change in NO_x emissions with manifold inlet temperature. Assume change in ambient is equal to change in manifold temperature. From Reference 41.

Figure 4-3. Effect of ambient temperature on NO_x emission levels from IC engines.[a]

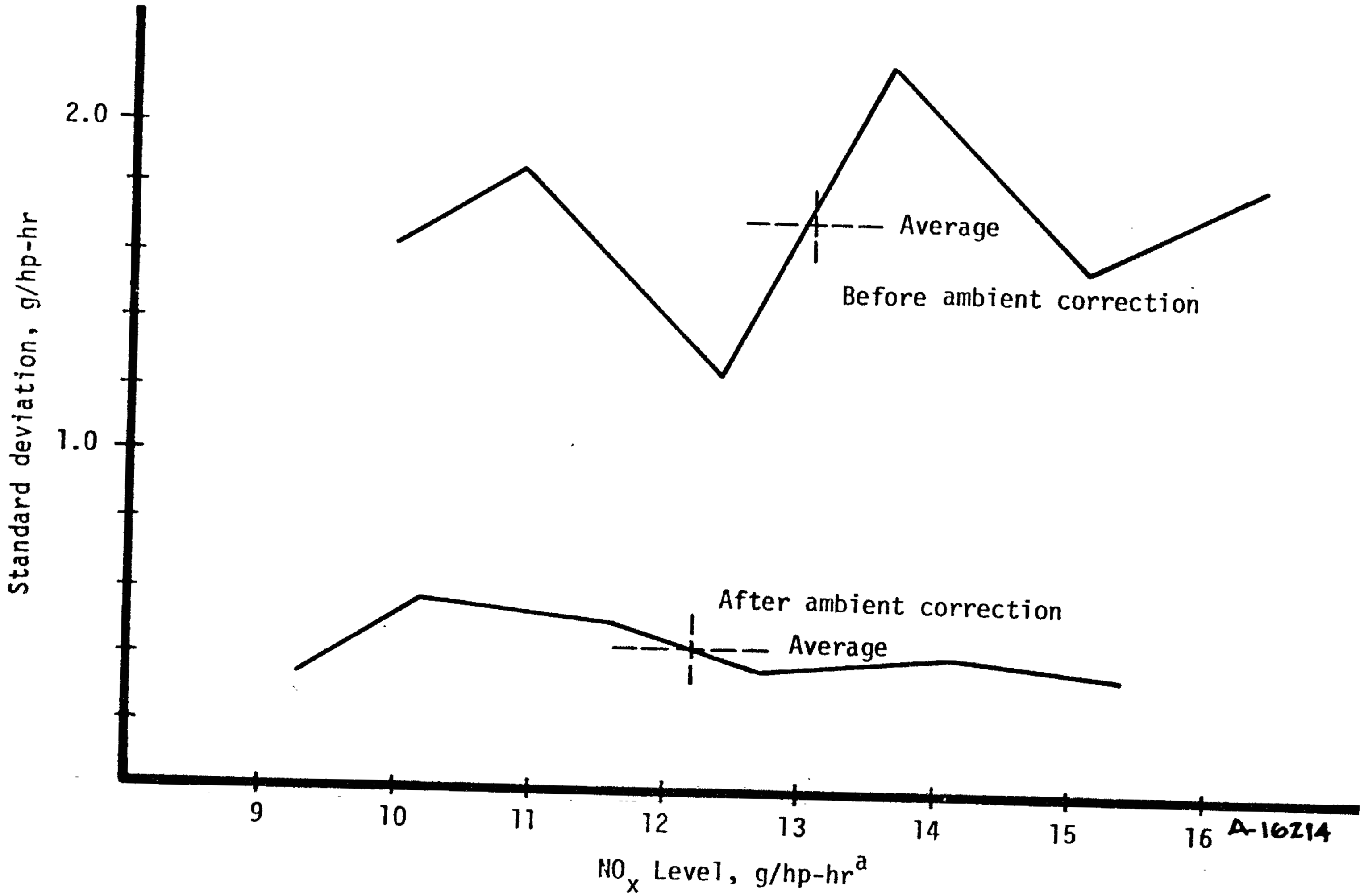
These changes were attributed largely to variations in A/F ratio in the carbureted gasoline engines. However, a correction factor for this effect could not be derived for the different carbureted engines because each engine's emissions response to changes in barometric pressure was too inconsistant to generalize a correction.

Despite the lack of a quantifiable correction for changes in ambient pressure, several studies have shown that scatter in emissions data taken at different ambient humidities and temperatures can be reduced, significantly, by applying the appropriate correction factors. For example, Figure 4-4(45) illustrates the reduction in data scatter that is achieved with emissions from HD gasoline engines by correcting for humidity only. The average standard deviation from the six engines was reduced by a factor of about six after applying a correction for humidity.

Figure 4-5(46) illustrates a similar result for ambient temperature and humidity correction of diesel engines emissions. The average scatter after correction was reduced to about one third of the scatter before correction. Note, that in absolute terms, the reduction in scatter for diesel engines was smaller (approx. 0.5 g/hp-hr) compared to the reduction for gasoline engines (approx. 1.5 g/hp-hr). Both studies, nevertheless, indicate that scatter in emissions data can be reduced significantly by correcting observed NO_x levels to standard conditions.

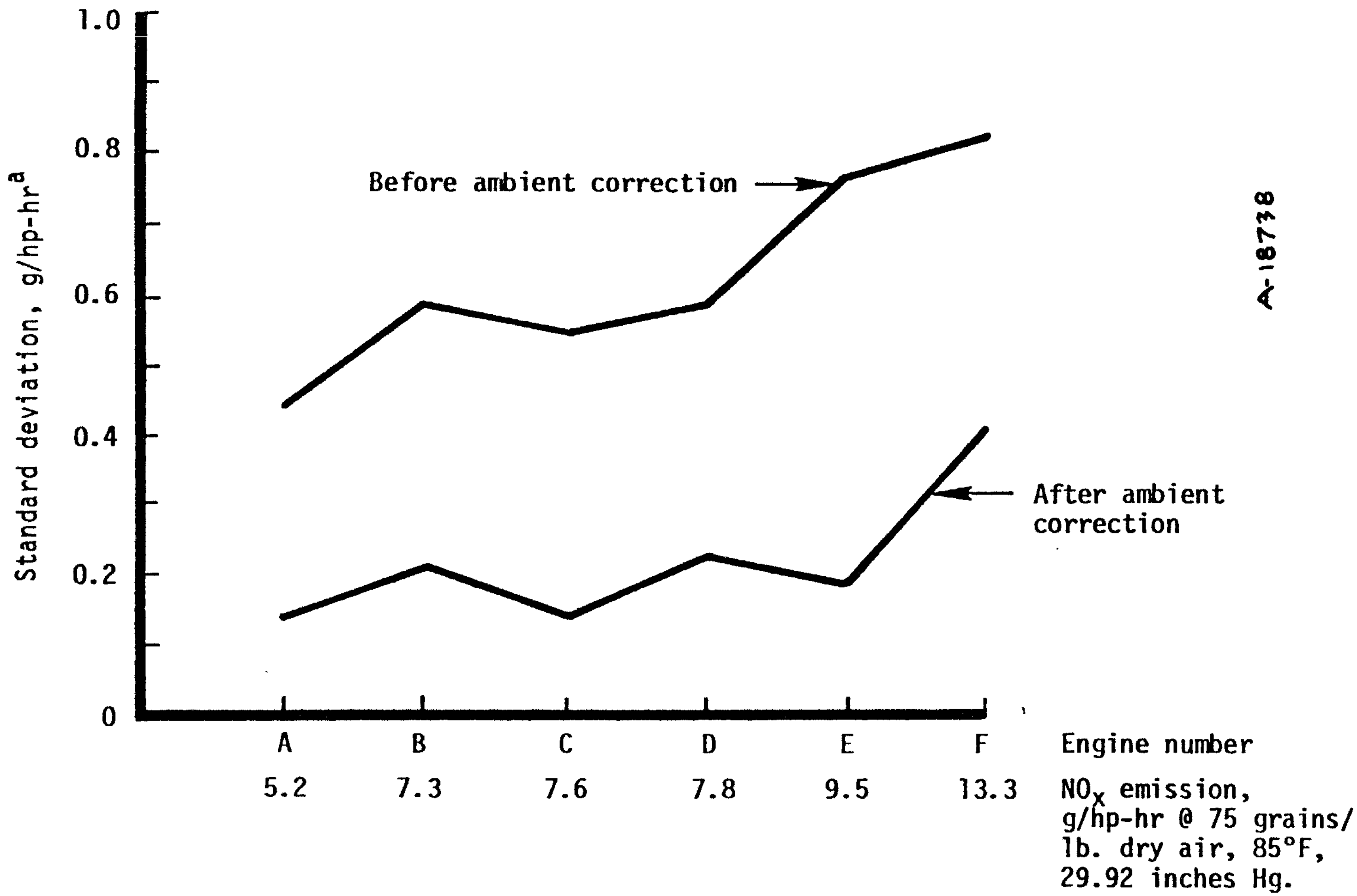
4.2.1.2 Selection of Existing Ambient Correction Factors for Application to Large-Bore IC Engines

All existing ambient correction factors were reviewed that potentially could be applied to adjust data from large-bore engines. A detailed description of this review is presented in Appendix C.2. Candidate factors are presented below. This discussion is divided into two sections depending



[a]Average level (Federal Test Cycle Composite) for four ambient humidities; 30, 45, 75, 120 grains/lb dry air.

Figure 4-4. Effect of humidity on emissions scatter for six HD gasoline engines (Reference 45).



[a]For Federal 13 mode composite cycle for HD diesel engines.

[b]Ambient humidity varied from 35 to 125 grains/lb dry; temperature from 70 to 115°F.

Figure 4-5. Effect of humidity and temperature on emissions scatter for six HD diesel engines[b] (Reference 46).

on whether the correction factor can be applied to data from (1) spark ignition (SI) or (2) compression ignition (CI) engines. Factors that have been selected to correct both uncontrolled and controlled emission data from the nine large-bore engine manfacturers are summarized following this discussion.

Factors Applicable to SI Engines

Three ambient humidity correction factors are potentially applicable to large bore, natural gas fueled engines, particularly four-stroke, carbureted versions. These factors are summarized in Table 4-1(47,48,49) Figure 4-6 is a comparison of the three factors over a typical range of ambient humidities. Note that only one of the three factors is at a constant load factor (Equation (2b)); the others are based on composite test cycles for vehicles.

As Figure 4-6 indicates, there is a considerable difference in correction factor depending on the study. All of the studies show, nevertheless, that ambient humidity has a significant effect on NO_x level. The result for Equation (3), based on light-duty automotive gasoline vehicle, shows the greatest sensitivity to variations in ambient humidity. These varied responses of NO_x level to changes in ambient humidity are not unexpected since engines react differently to changes in inlet conditions. Their response generally depends on their A/F ratio, fuel metering and distribution system, and ignition characteristics. Since large-bore IC engines typically operate at a constant rated load, the constant load correction factor (Equation (2b)) has been selected for application to the reported data.

TABLE 4-1. AMBIENT HUMIDITY CORRECTION FACTORS FOR SI ENGINES

Equation No.	Correction Factor[a]	Reference
(1)	$K = 0.634 + 0.00654(H) - 0.0000222(H)^2$ Composite Factor (9 Mode Federal HD Gasoline Test Cycle)	47
(2a)	$K = 0.796 + 0.175(H/100) + 0.129(H/100^2)$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles)	48
(2b)	$K = 0.844 + 0.151(H/100) + 0.075(H/100)^2$ 50 mph, Constant Load	48
(3)	$K = 1/(1 - 0.0047(H-75))$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles)	49

[a] H = specific humidity in grains H_2O/lb dry air.

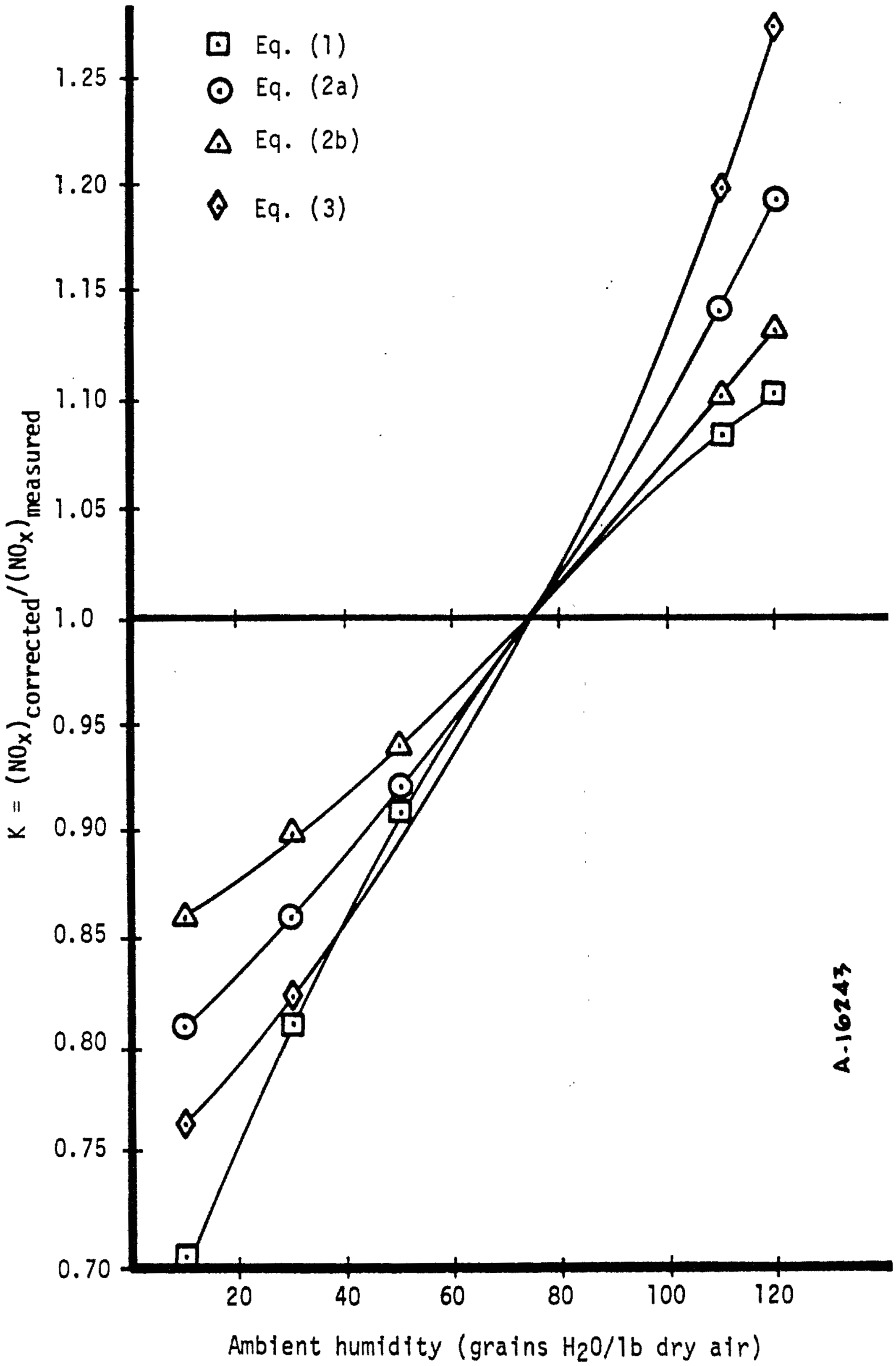


Figure 4-6. Comparison of SI ambient humidity correction factors.

Previous investigators have been unable to establish an ambient temperature correction factor for spark ignition IC engines because various automotive engines respond quite differently to inlet air temperature variations (see Appendix C.2). A limited amount of data exists for large IC engines that show the variation in NO_x emissions with ambient temperature, or manifold air temperature for turbocharged units. Figure 4-7[50] illustrates the change in NO_x level with a change in manifold air temperature for a large-bore, four stroke per cycle, turbocharged (4-TC) gas engine. This response indicates approximately a 1-percent change in NO_x level per ^oF change in manifold air temperature. (A change in manifold air temperature is nearly equivalent to the same change in ambient temperature.)

Figure 4-8[51] indicates the response of NO_x emissions with changes in ambient air temperature for large-bore, blower-scavenged gas engines. These results show approximately a 2-percent change in NO_x per ^oF change. Both Figures 4-7 and 4-8 indicate that NO_x emissions from large-bore engines are very sensitive to ambient temperature variations. Therefore the results of Figure 4-7 will be applied to all turbocharged gas engine data, and the results of Figure 4-8 will be applied to all nonturbocharged gas engine data.

Factors Applicable to IC Engines

A survey of the literature established two sources that have reported ambient correction factors for truck-size diesel engines. In the study by Krause, et. al., a factor was developed that included the effects of temperature and humidity[52]. The results of this study were subsequently adopted by the EPA for mobile heavy duty diesel engines. The other study was conducted by the Coordinating Research Council (CRC) and only investigated

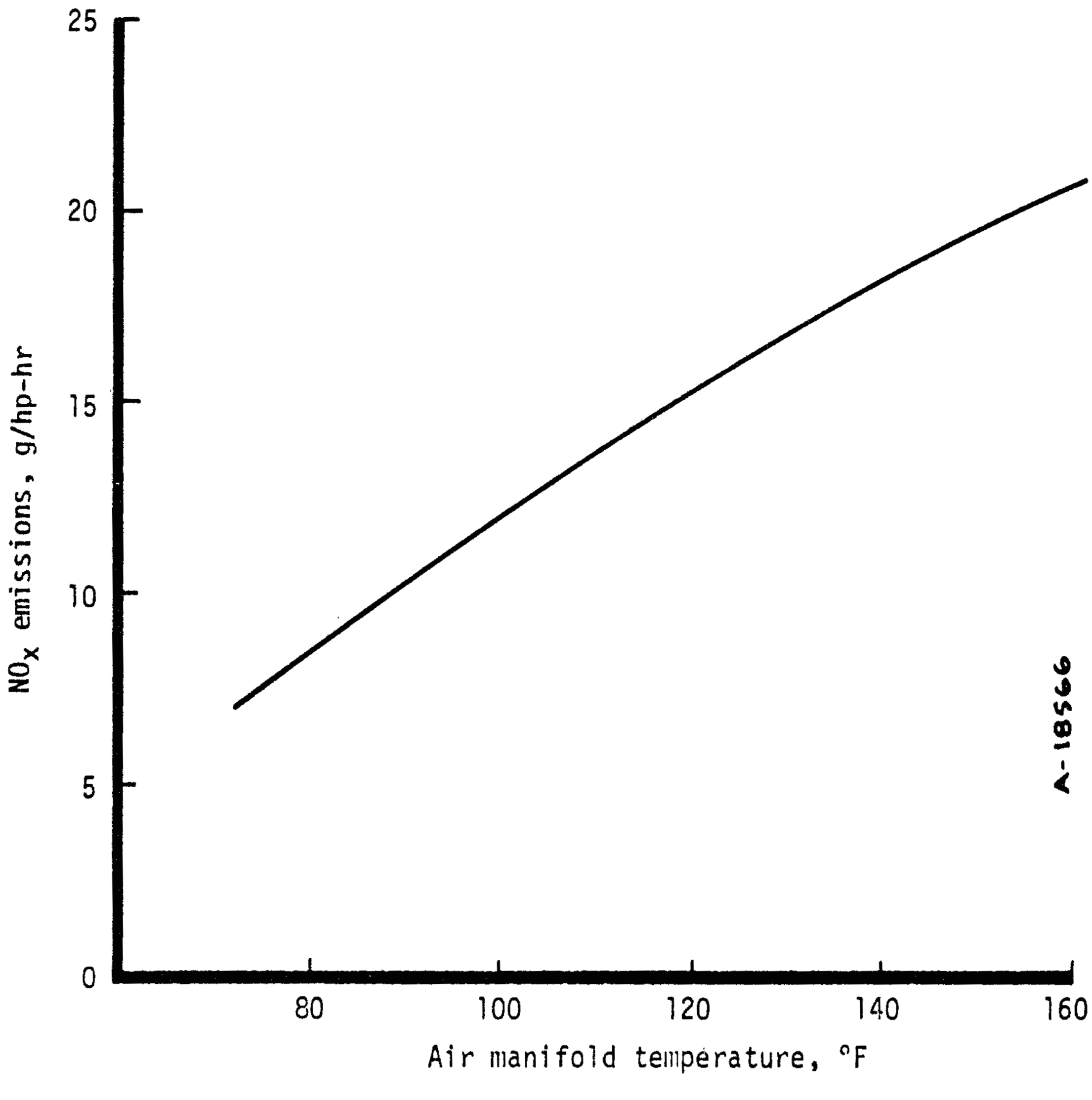


Figure 4-7. Effect of manifold air temperature on a large-bore 4-TC engine (Reference 50).

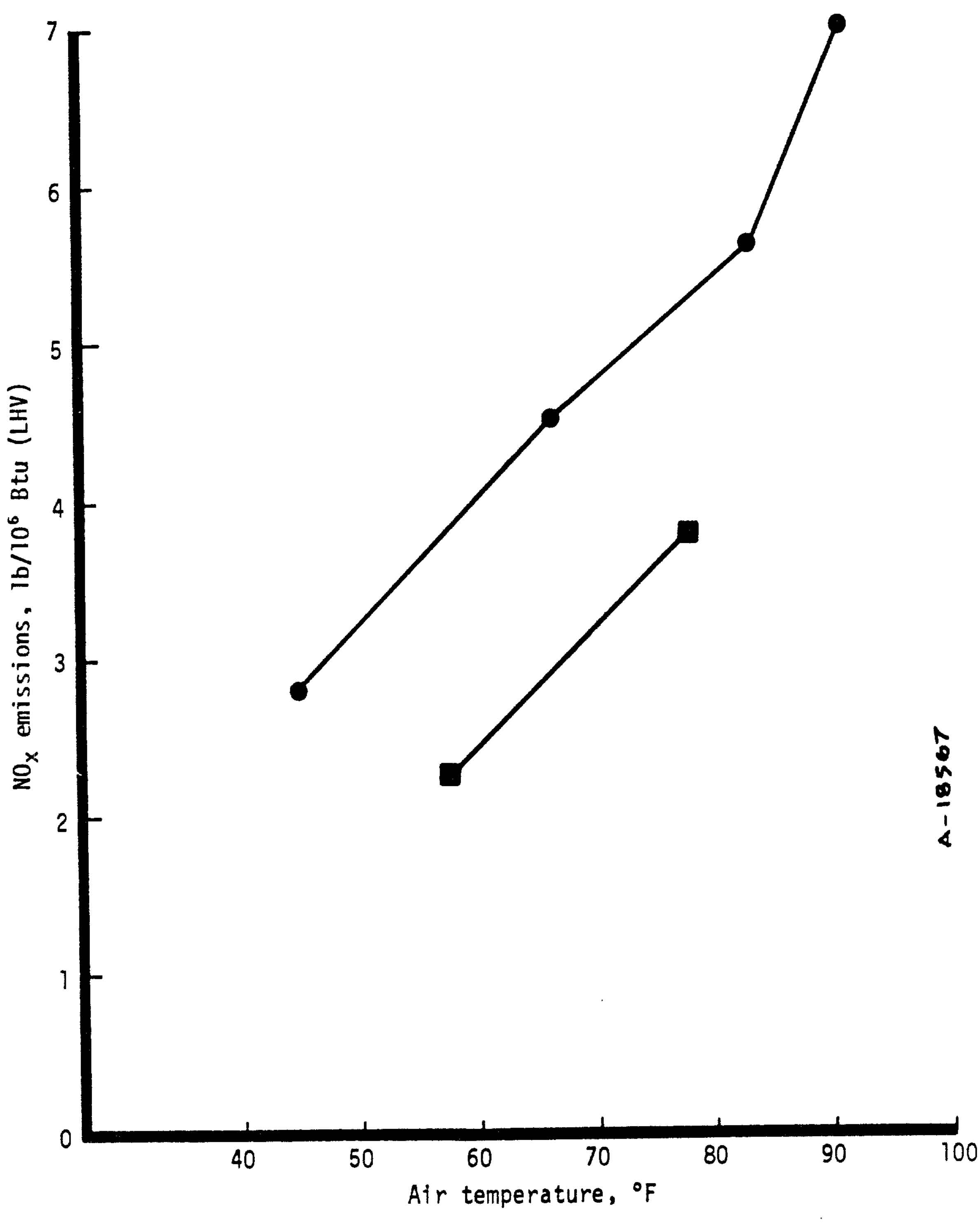


Figure 4-8. Test results of NO_x emissions versus intake air temperature for two blower-scavenged gas engines (Reference 51).

the effects of ambient humidity[53]. (A more detailed discussion of both studies can be found in Appendix C.2.)

Figure 4-9[54,55] shows the ambient humidity correction factors developed from these two studies. The ambient humidity factors for SI engines are also shown in Figure 4-9. In general, SI engines appear to be more sensitive to ambient humidity variations than CI engines. Note that the results from the two CI engine studies are shown for specific CI engine types (e.g., four-stroke turbocharged, aftercooled engines). In general, the data show that NO_x emissions from different engine types, particularly at low humidity levels, respond differently to changes in ambient humidity.

The Krause study also investigated the effect of ambient temperature on NO_x emissions. Figure 4-10[56] presents the correction factors that were derived for these smaller bore engines. It shows that NO_x emissions from naturally aspirated and blower-scavenged engines are more sensitive to inlet air temperature changes than are the emissions from aftercooled units.

Since the Krause study systematically examined the effect of both temperature and humidity for a number of CI engine types, his correction factors have been selected for application to similar large-bore engine types.

4.2.1.3 Summary of Ambient Correction Factors for Application to Large-Bore Engine Data

Table 4-2[57,58,59] summarizes the ambient correction factors that have been selected for application to the data reported by the nine large-bore engine manufacturers. Note that, with the exception of SI temperature factors, all the corrections are based on studies of smaller bore automotive engine types. The corrections for CI engines are given for specific engine

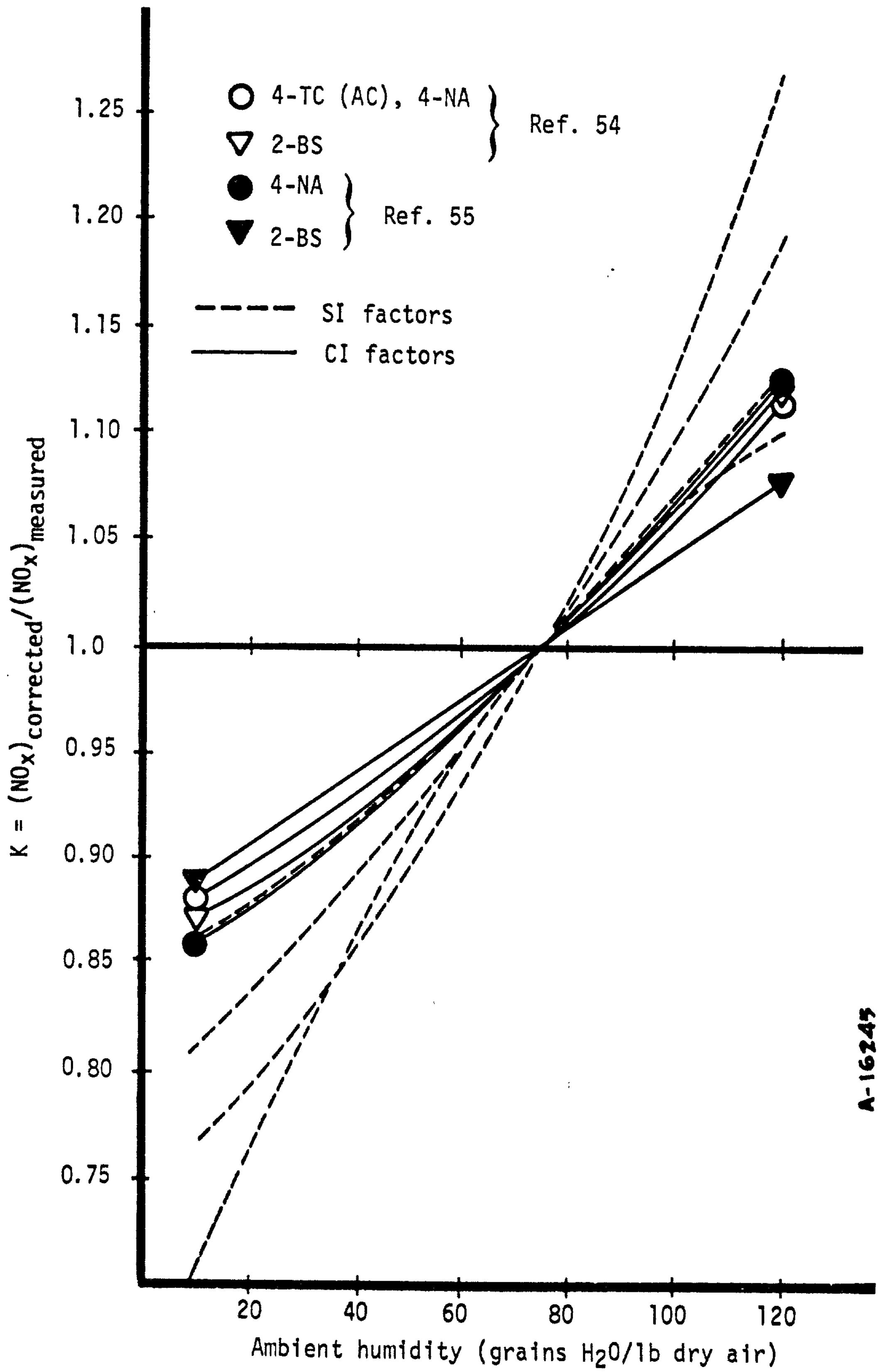


Figure 4-9. Comparison of CI and SI ambient humidity correction factors.

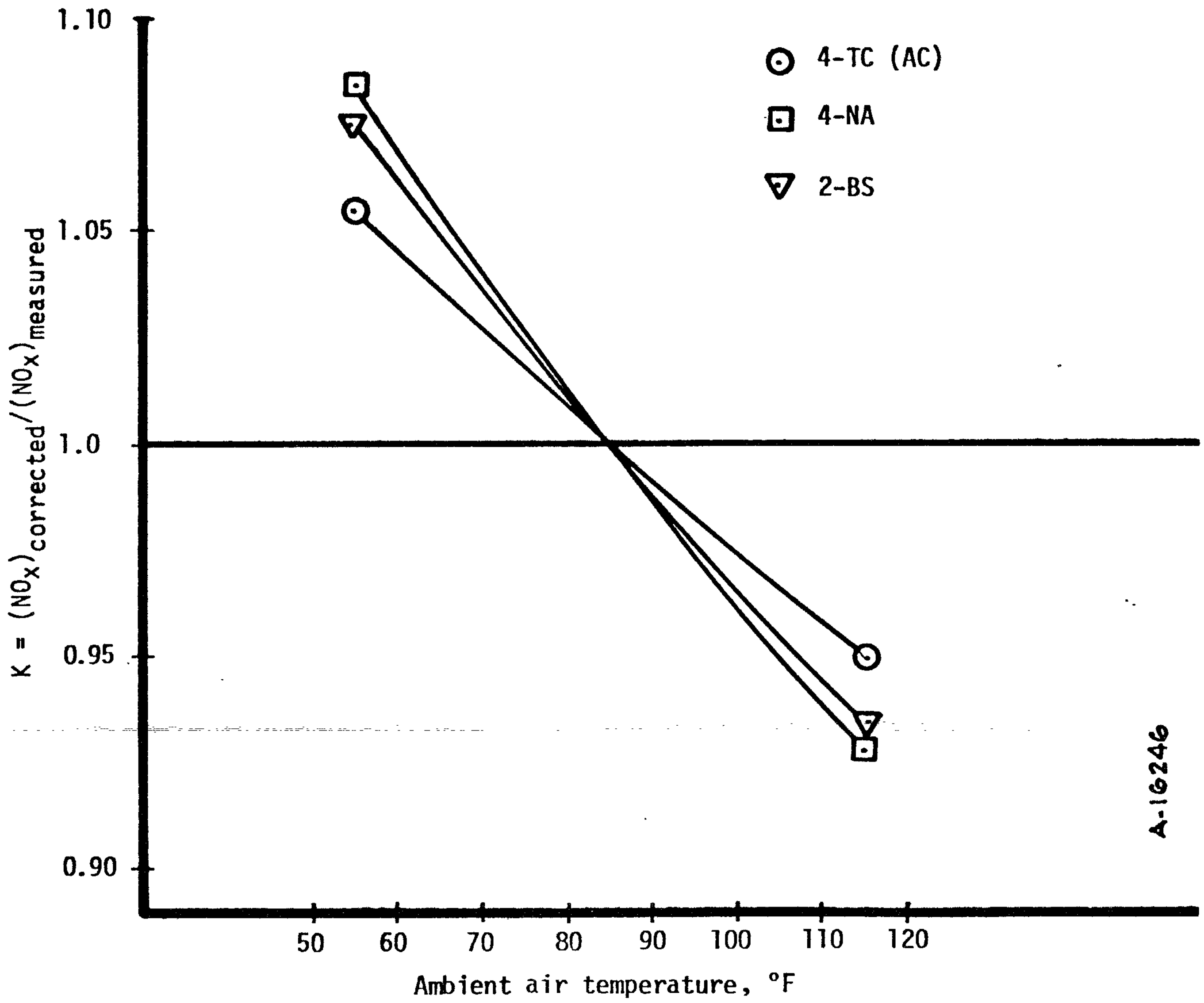


Figure 4-10. Correction factors for temperature for CI engines (Reference 56).

TABLE 4-2. EXISTING IC ENGINE AMBIENT CORRECTION FACTORS FOR APPLICATION TO LARGE-BORE ENGINES

Fuel	Correction Factor[a]			Comments
Diesel & Dual Fuel (CI) Humidity and Temperature Correction	$K = 1/(1+A(H-75) + B(T-85))$ H = observed humidity, grains H_2O/lb dry air T = observed inlet, air temperature, °F			Rated load correction for humidity and temperature (from Ref. 57).
	Type	A	B	
	2-BS	-0.00242	0.00235	
	4-NA	-0.00231	0.00255	
	4-TC, AC	-0.00231	0.0017	
Natural Gas (SI)				
Humidity	$K = 0.844 + 0.151 \left(\frac{H}{100}\right) + 0.075 \left(\frac{H}{100}\right)^2$			Rated load correction for humidity only (from Ref. 58)
Temperature	BS $K = 1 - (T - 85)(0.017)$ TC $K = 1 - (T - 85)(0.010)$			Ambient temperature correction only (from Ref. 59)

[a] NO_x corrected = (K) NO_x observed

types. In addition all the factors are based on rated load conditions, since large bore engines typically operate at, or near, rated load.

Although ambient humidity and temperature variations can significantly affect the NO_x emissions that are measured from a particular engine, these variations, in general, are not responsible for the large variations in uncontrolled emissions that were reported for similar engine types by different engine manufacturers. The other sources of data variability (largely measurement practices and design differences among models) are discussed in Sections 4.2.2 and 4.3.

4.2.2 Effect of Measurement Practices

Previous studies have shown that sampling instrumentation and procedures have a large effect on emission levels. For example, a series of studies conducted by the Coordinating Research Council (CRC) indicated that uncertainties in the measurement of NO_x levels can range as high as 40 percent[60]. This conclusion was based on the standard deviation of measurements reported by different laboratories for the same emission source, expressed as a percentage of the mean emission level. CRC concluded that this uncertainty could be attributed largely to poor calibration and measurement procedures. The EPA, then in cooperation with CRC, showed that these uncertainties could be reduced to less than 5 percent using a specific set of procedures[61]. Since then, EPA has proposed that these procedures be used to certify mobile, heavy-duty diesel and gasoline engines starting with the 1979 model year. The following paragraphs will briefly discuss the measurement practices of each of the nine large-bore engine manufacturers who reported emissions data. (Additional details regarding these practices can be found in Appendix C.3.) Then uncertainties for each manufacturer's practice will be estimated relative to the proposed EPA procedure. These uncertainties will be used in Section 4.3 to establish upper and lower bounds for estimated average emission levels from large-bore IC engines.

Table 4-3[62,63,64,65] indicates which of the four measurement practices was used by each of the manufacturers. Note that two manufacturers, Alco and Ingersoll-Rand, used what is essentially the EPA procedure. For the purpose of this discussion the EPA procedure will serve as a reference for comparison with the other three procedures, since it is believed to be the most accurate.

Figure 4-11 illustrates each of the four procedures schematically, and Table 4-4 summarizes the sources of error for the DEMA, SAE, and EMD practices. (A more detailed discussion of these procedures can be found in Appendix C.3.) The primary shortcoming of these practices is their failure to adequately define instrument performance and sample transfer procedures. Unheated sample lines, inappropriate water removal devices, system leaks, and failures of the convertor in the chemiluminescent instrument all lead to errors in the measurement of NO_x in the sample gas. The use of NDIR instruments (SAE/EMD practices), can lead to overstated values of NO_x emissions due to interferences resulting from the presence of water vapor in the detector cell of the NDIR instrument. Considered together, these sources of error can cause a large uncertainty in reported NO_x levels. Figure 4-12 illustrates the overall uncertainty for data reported by each of the nine engine manufacturers. (Again, a more detailed discussion of these uncertainties can be found in Appendix C.3.) Note that manufacturers using the SAE or EMD procedure could experience uncertainties of $\pm$20 percent. Manufacturers using the DEMA practice, in contrast, are more likely to experience understated (5 to 15 percent) NO_x levels due to a loss of NO_x

TABLE 4-3. LARGE-BORE ENGINE MANUFACTURERS MEASUREMENT PRACTICES

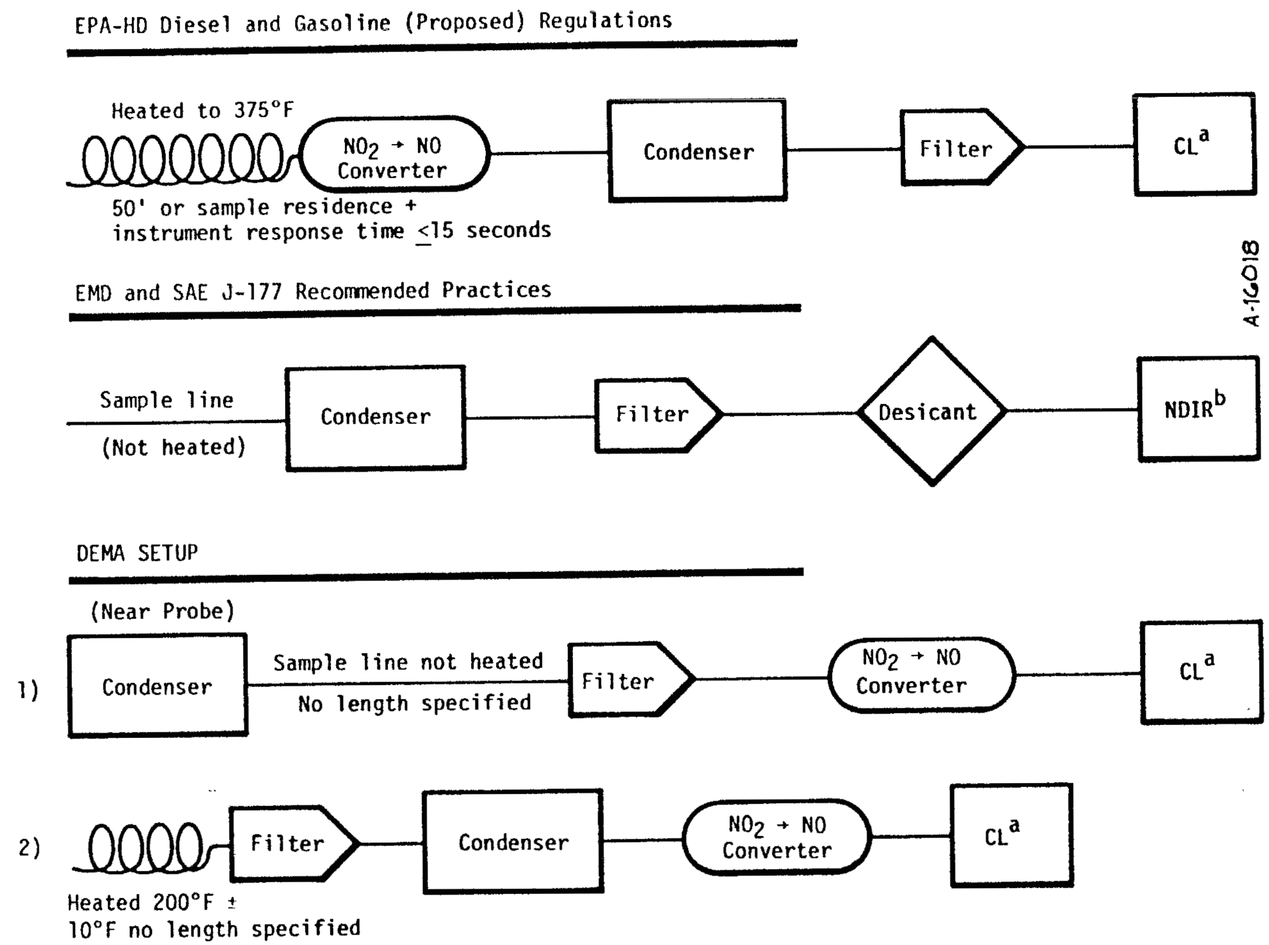
Manufacturer	Measurement Practice EPA[a]	DEMA[b]	SAE[c]	EMD[d]
Alco	X			
Caterpillar			X	
Colt		X		
Cooper Energy		X	X	
DeLaval		X		
ElectroMotive (GMC)				X
Ingersoll-Rand	X			
Waukesha			X	
White Superior (Div. Cooper)		X		

[a]EPA's proposed practice for 1979 Heavy Duty Diesel and Gasoline Engines(62).

[b]Diesel Engine Manufacturers Association (DEMA) Exhaust Emission Measurement Procedure for Low and Medium Speed Engines(63).

[c]Society of Automotive Engineers (SAE) Recommended Practice J177a(63).

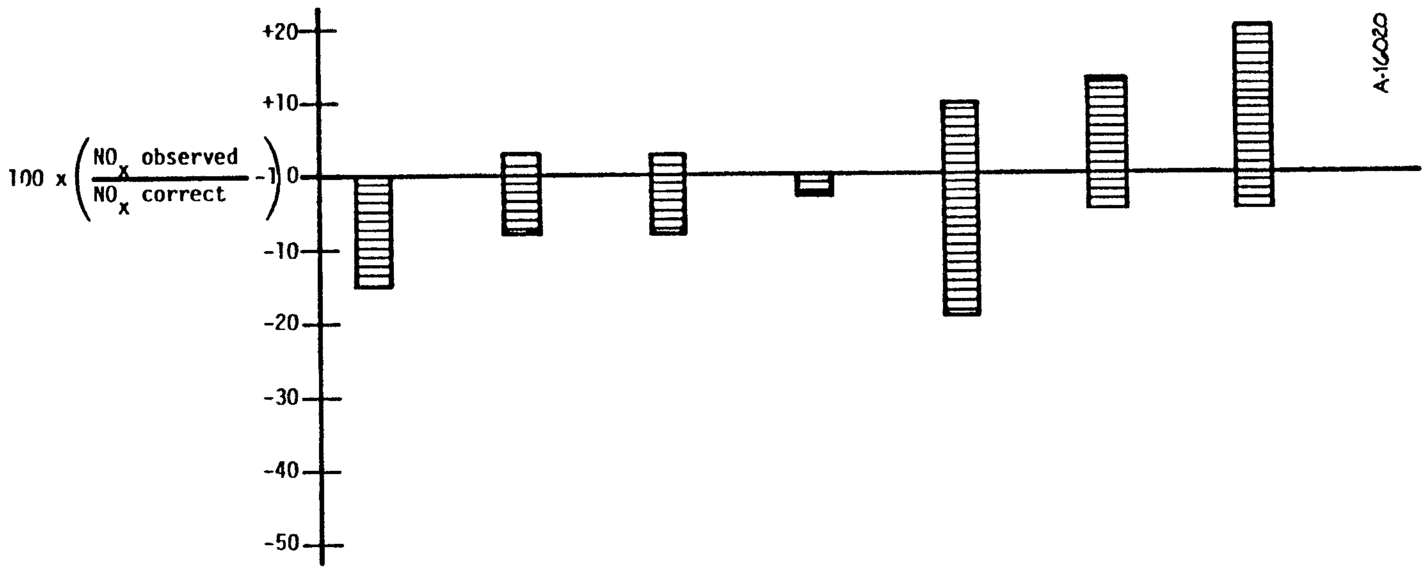
[d]ElectroMotive Division of General Motors Corporation Practice(65).



[a]CL — Chemiluminescent analyzer

[b]NDIR — Nondispersive infrared analyzer

Figure 4-11. Sampling practices.



Mfg	Colt	Cooper	Delaval	White Superior	Waukesha	Cooper	GMC/EMD
Practice	DEMA				SAE		EMD

[a]Does not include potential error due to system leaks, and assumes chemical drier is conditioned.

Figure 4-12. Measurement uncertainty relative to EPA procedure.[a]

TABLE 4-4. SOURCES OF ERROR FOR DEMA, SAE, EMD EMISSION PRACTICES

DEMA (CL)	EMD (NDIR/NDUV)/SAE (NDIR)
• Unheated sampling lines permitted • No specification of sample residence time in sampling line, or system response time • Leak checks are not specified • Water removal device can be located at analyzer, but before $NO_2 \rightarrow NO$ absorption in water trap • Chemical driers permitted • No instrument specifications • Converter efficiency checks not specified • No calibration procedures specifed • Calibration and span gas specifications not defined	• Unheated sample lines • No sample residence or system response time specified • Leak checks not specified • Allow chemical drier • Calibration procedures not specific (e.g., what constitutes out-of-calibration, how calibration points are curve fit, etc.) • Calibration and span gas blends and dilutents not specified by SAE. EMD has "own" specifications.

during the transfer of the sample gas to the relatively interference-free chemiluminescent analyzer. These estimates of uncertainty will be used to place upper and lower bounds on the average uncontrolled emission levels computed in Section 4.3.

4.3 UNCONTROLLED EMISSION LEVELS

This section presents data on uncontrolled emissions from large-bore engines. Average uncontrolled NO_x emissions from these engines, weighted according to sales, were derived from data supplied by manufacturers. By applying a specified degree of NO_x control to these average uncontrolled emission levels, potential controlled (regulated) emission levels can be established. The degrees of NO_x control that can be applied are identified in Chapter 6, which summarizes demonstrated alternative controls.

This approach to setting the standards requires an adequate sample of emission data for each manufacturer's engine. Section 4.3.1 discusses the current data base, and shows that emissions data have been reported for about 80 percent of all the large-bore engine models manufactured to burn diesel, dual fuel, and natural gas. This large existing data base is representative of all the engines to be affected by standards of performance.

Section 4.3.2 presents the uncontrolled emissions data for diesel, dual fuel, and natural gas engines, and examines the sources of variations in these data. Differences in the ambient conditions (temperature and humidity) and procedures for measuring the emissions account for only small variations in the data. The largest source of data variations is differences in engine design. These differences and their effect on NO_x emission levels are discussed in Section 4.3.3.

Because of these differences, a method is needed to characterize uncontrolled emissions from each of the three fuels for which standards of performance will be proposed. In Section 4.3.4, representative uncontrolled NO_x levels for each fuel are determined by weighting each manufacturer's data (corrected for ambients where possible). Weighting is based on the percentage of total horsepower sold by each mnaufacturer during the past 5 years, and the weighted levels are bounded by estimates of measurement uncertainty, based on each manufacturer's procedures.

4.3.1 Existing Data Base

The extent of the current emissions data base is illustrated in Table 4-5, which shows the number of large-bore manufacturers who produce engine models within the diesel, dual fuel, and natural gas categories. The second row of the table shows the number of models produced for each engine type (e.g., 2-BS, 2-TC, 4-NA, and 4-TC). The lower two rows show the number of models within each fuel and engine type category that contain (1) uncontrolled and (2) controlled emissions data. Uncontrolled emissions data are available from every manufacturer of large-bore engines, although a few manufacturers have not conducted tests to reduce NO_x emissions from their engines.

In general, as shown in the last column of Table 4-5, there are uncontrolled emissions data for about 80 percent of the models produced for each fuel category. The current data base contains more data than for those models listed in Table 4-5, since there are data for several engines of the same model for some manufacturers. This additional data is useful in determining differences in emissions data from engines of the same, and different manufacturers, as discussed in Section 4.3.3. Thus, a substantial data base exists for characterizing uncontrolled emissions from diesel, dual fuel, and natural gas engines.

TABLE 4-5. EXTENT OF EXISTING DATA BASE

Fuel	Diesel				Dual Fuel		Gas				Overall		
Strokes	2		4		2	4	2		4				
Air Charging	BS	TC	NA	TC	TC	TC	BS	TC	NA	TC	Diesel	Dual Fuel	Gas
No. of manufacturers	2	2	1	5	1	4	2	2	4	6	6	4	7
No. of models[a]	2	2	3	8	1	5	2	5	5	14	15	6	28
No. of models with uncontrolled data	2	2	1	6	1	4	2	6	4	10	11	5	23
No. of models with controlled data	2	2	1	5	1	3	2	6	3	7	10	4	18

[a]A model is a group of engines that share the same fuel, air charging, strokes/cycle, manufacturer, bore, and stroke.

4.3.2 Uncontrolled Emission Levels

Uncontrolled emissions of NO_x, CO, and HC (and nonmethane HC where measured) are shown on Figures 4-13 to 4-15. On each figure the data is plotted separately for each fuel (diesel, dual fuel, and gas), and is differentiated by engine type (i.e., 2-BS, 2-TC, 4-NA, and 4-TC). Since in general, the CO and nonmethane HC levels from these engines are considerably lower than the limits that apply to mobile vehicles and engines,[4/] this section is concerned primarily with NO_x emissions. The effects of NO_x reduction techniques on CO, HC, and smoke emissions are discussed in Section 4.4.12. Figures 4-13 through 4-15 show that uncontrolled emission levels vary considerably within each category of fuel and engine type. In Figure 4-13, fuel consumption, particularly for diesel and dual-fuel engines, remains relatively constant despite wide variations in NO_x levels among all engine types. Since both NO_x emissions and thermal efficiency increase as cylinder temperature increases, efficient engines (low fuel consumption) would be expected to show high NO_x emission rates. As shown in Figure 4-16, this is not the case among all engines of one fuel. NO_x levels and four- and two-stroke diesel engines shown on Figure 4-16(a), however, appear to increase as fuel consumption decreases, but other trends are not apparent for other fuels and engine types. Although other design features (e.g., manifold air temperature, air-to-fuel ratio, speed, torque, etc.) are probably more

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[4/] For example, the proposed Federal Government standards beginning in 1979 for heavy-duty gasoline and diesel engines are 1.5 g/hp-hr hydrocarbon (HC), 25 g/hp-hr carbon monoxide (CO), and 10 g/hp-hr hydrocarbon plus oxides of nitrogen (HC + NO_x). California regulations for 1977-1978 heavy duty vehicles (greater than 6,000 lbs) are 1.0 g/hp-hr HC, 25 g/hp-hr CO, and 7.5 g/hp-hr NO_x.

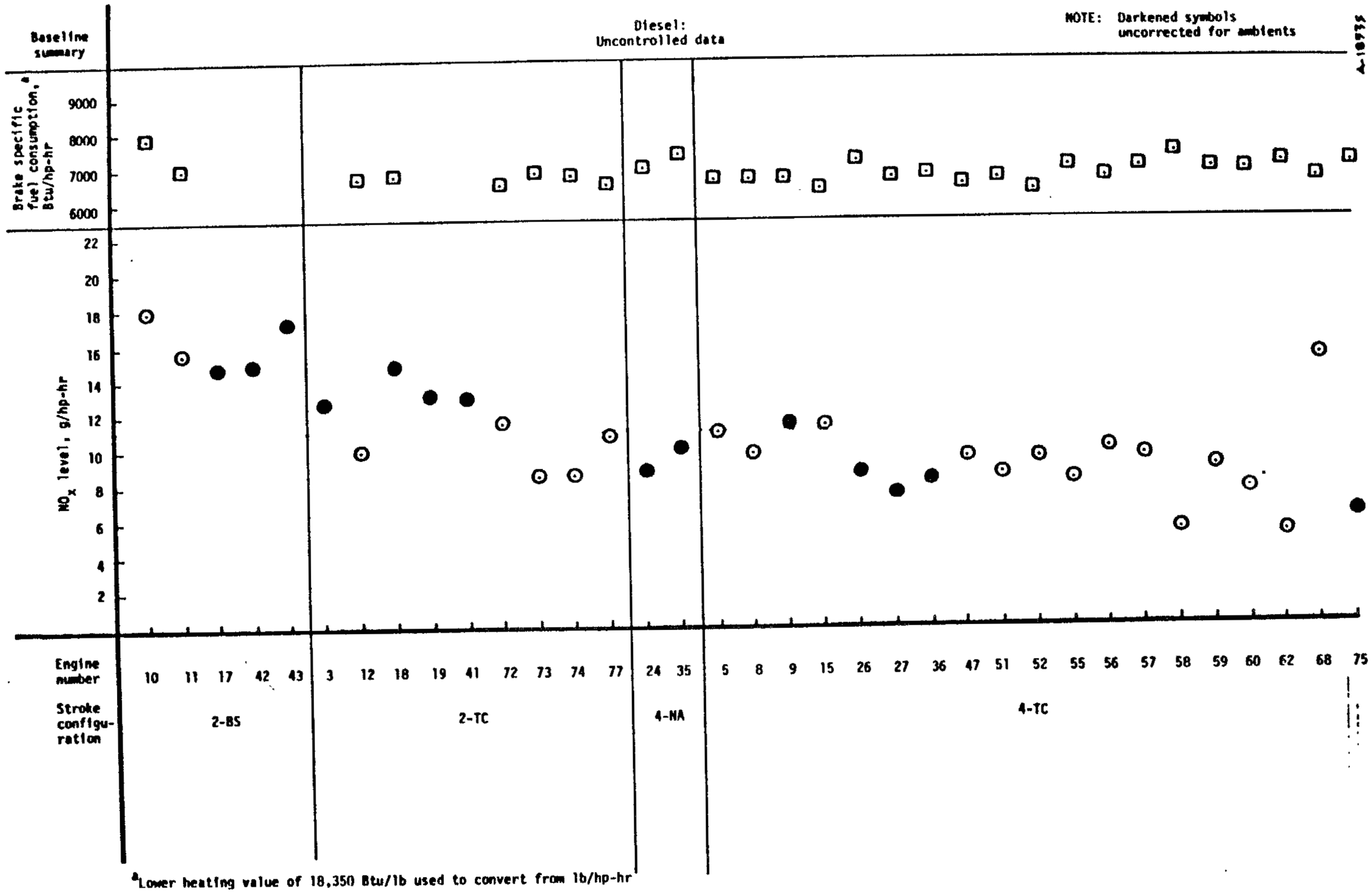


Figure 4-13(a). Uncontrolled NO_x emissions from diesel engines.

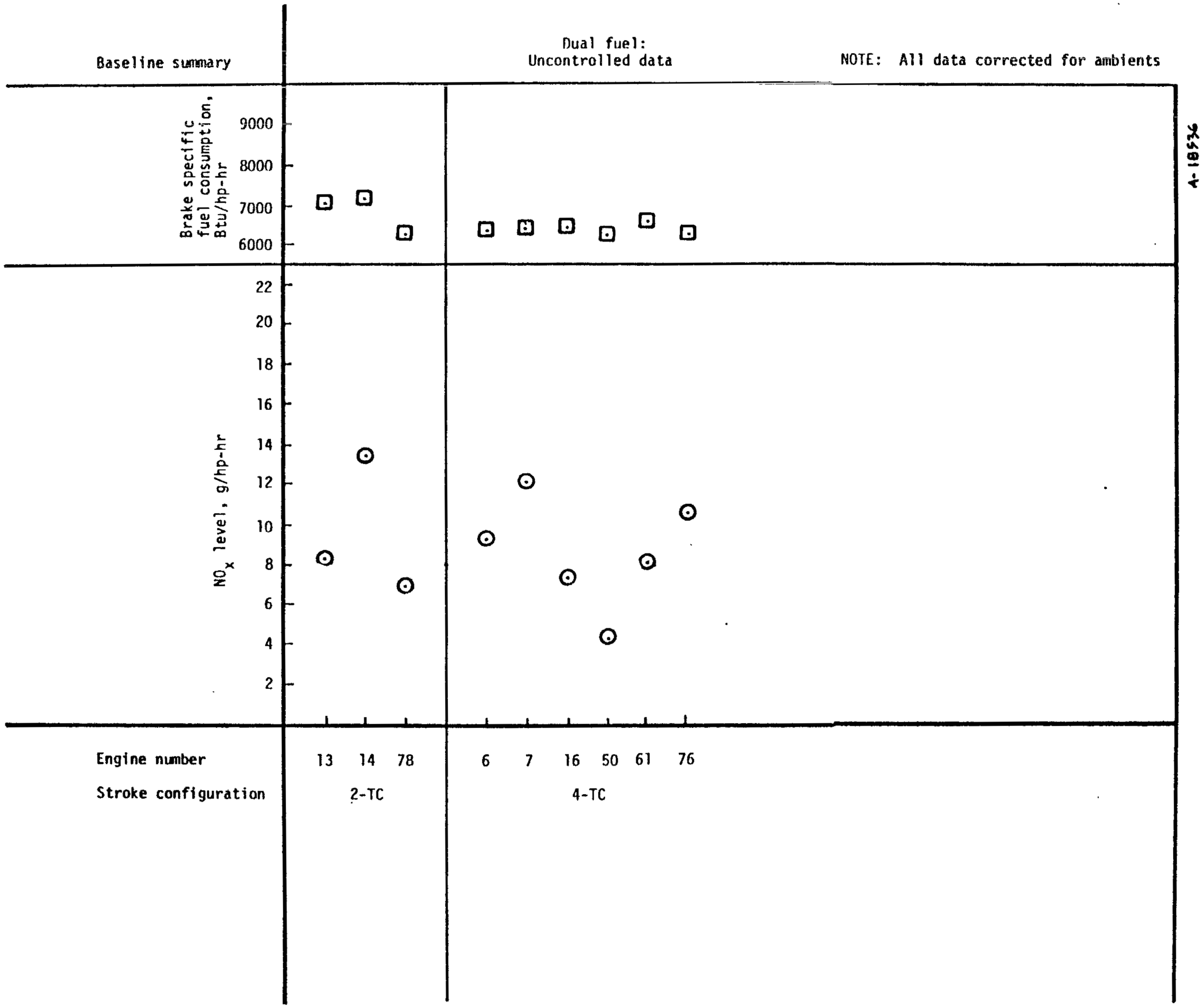


Figure 4-13(b). Uncontrolled NO_x emissions from dual fuel engines.

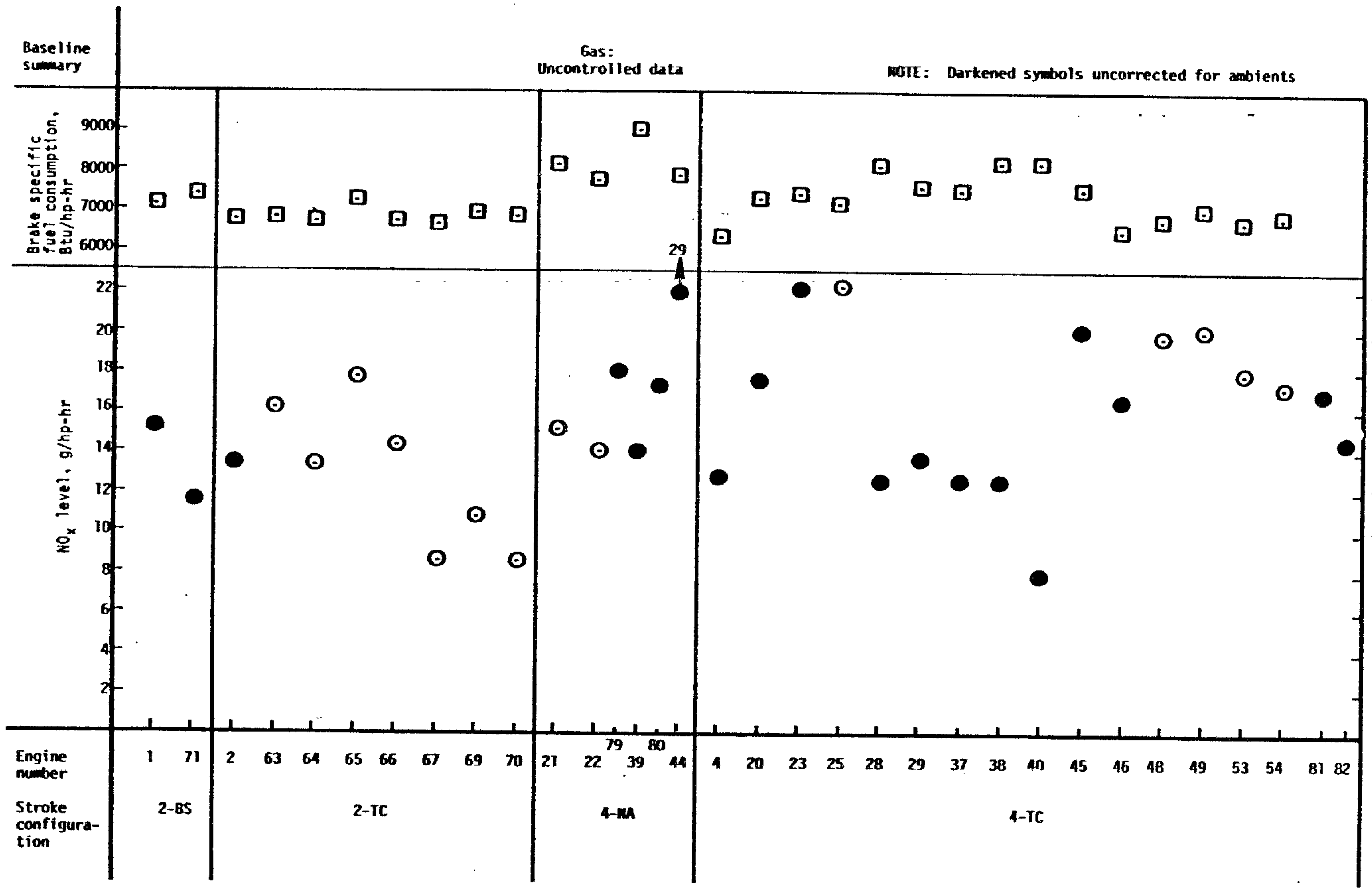


Figure 4-13(c). Uncontrolled NO_x emissions from gas engines.

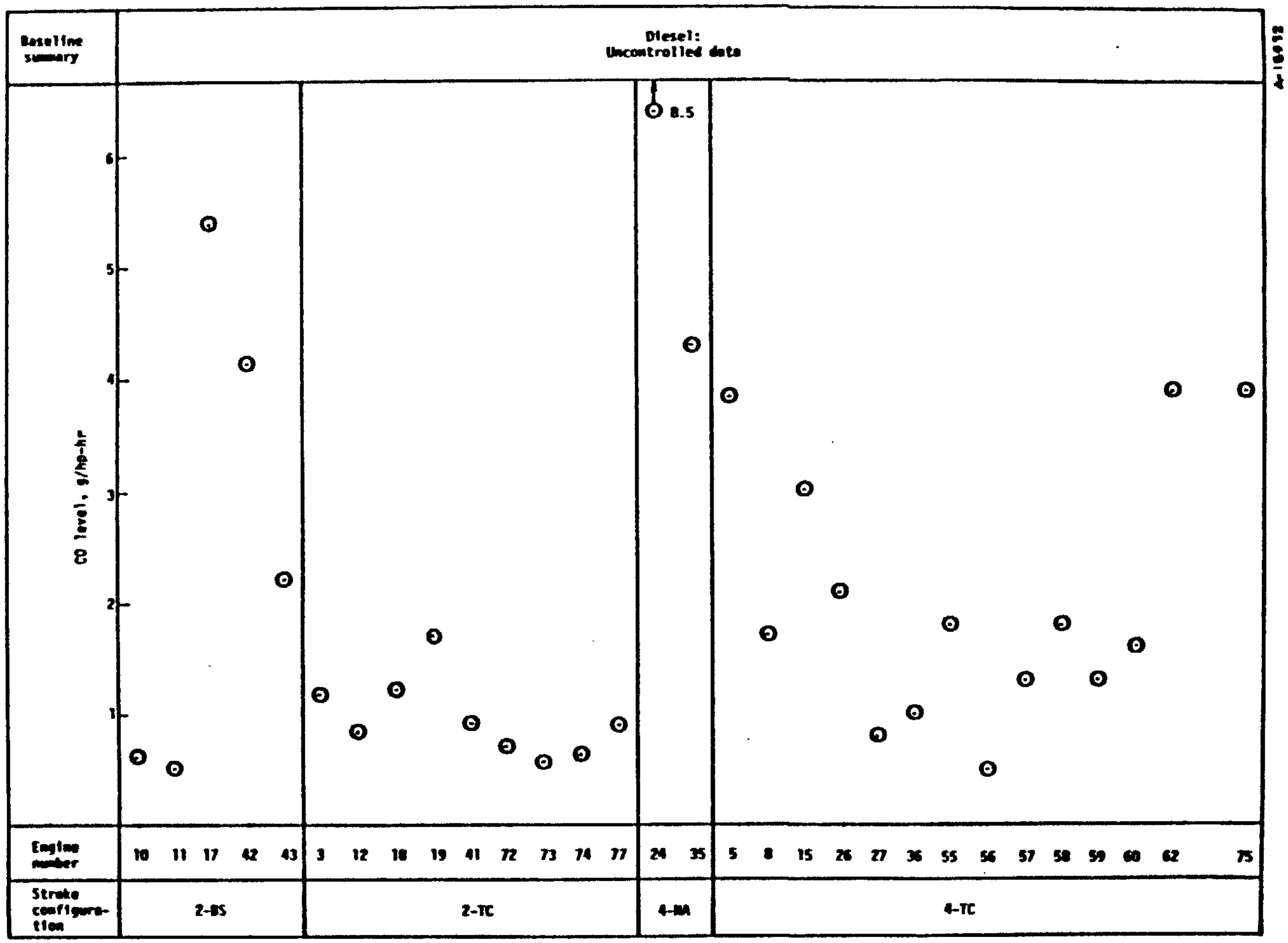


Figure 4-14(a). Uncontrolled CO emissions from diesel engines.

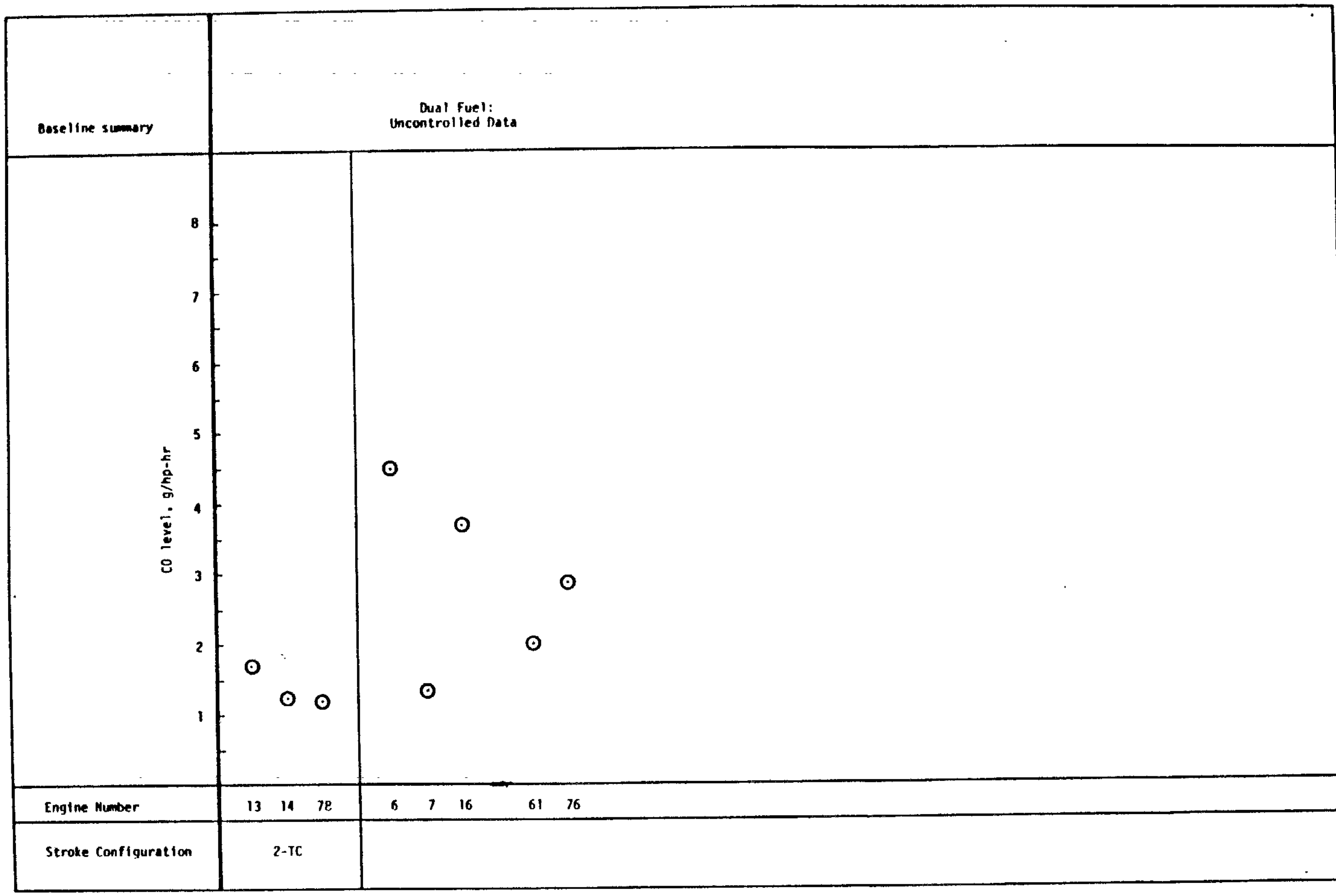


Figure 4-14(b). Uncontrolled CO emissions from dual fuel engines.

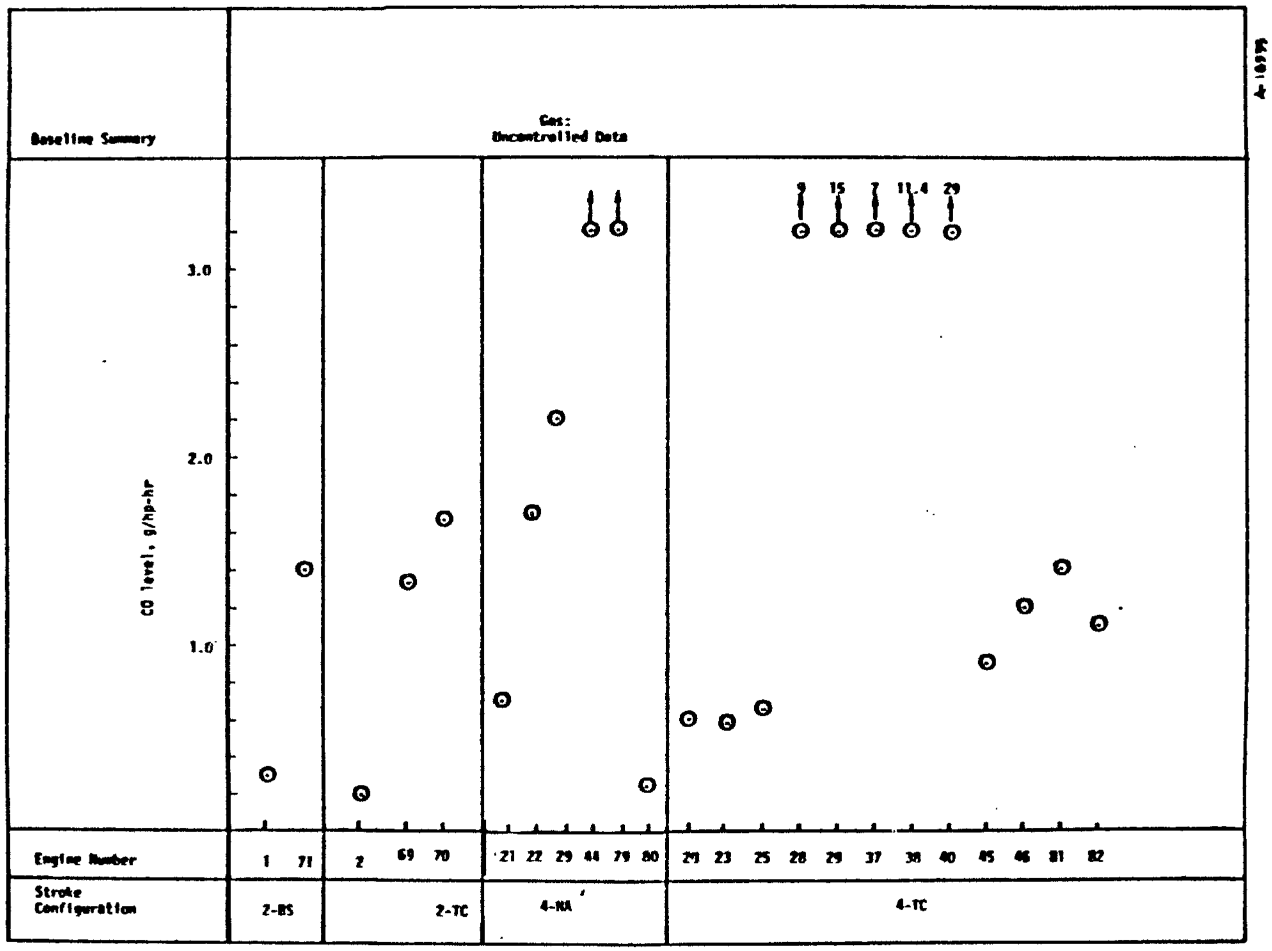
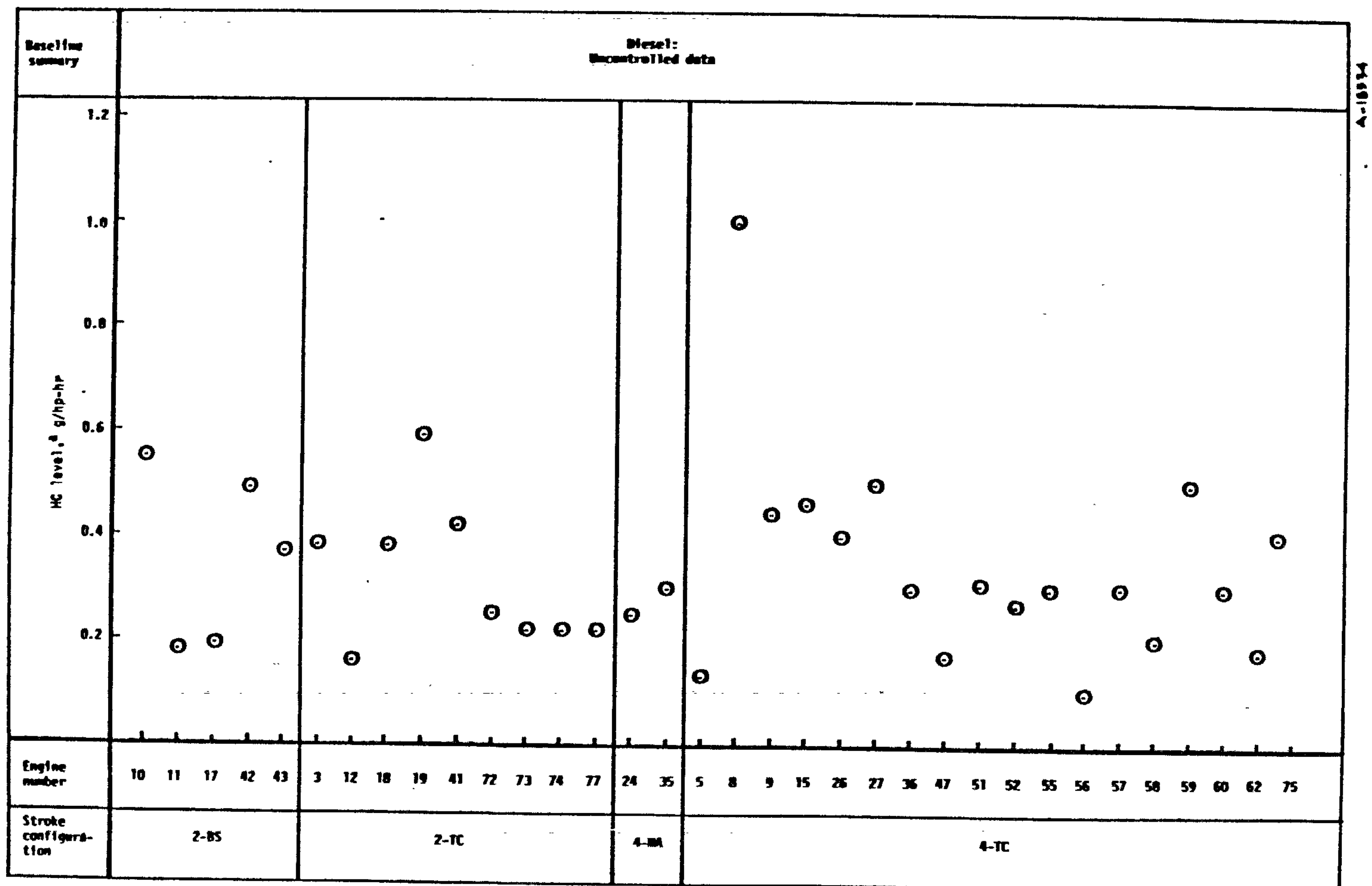


Figure 4-14(c). Uncontrolled CO emissions from gas engines.



[a]HC level assumed to be all nonmethane.

Figure 4-15(a). Uncontrolled HC emissions from diesel engines.

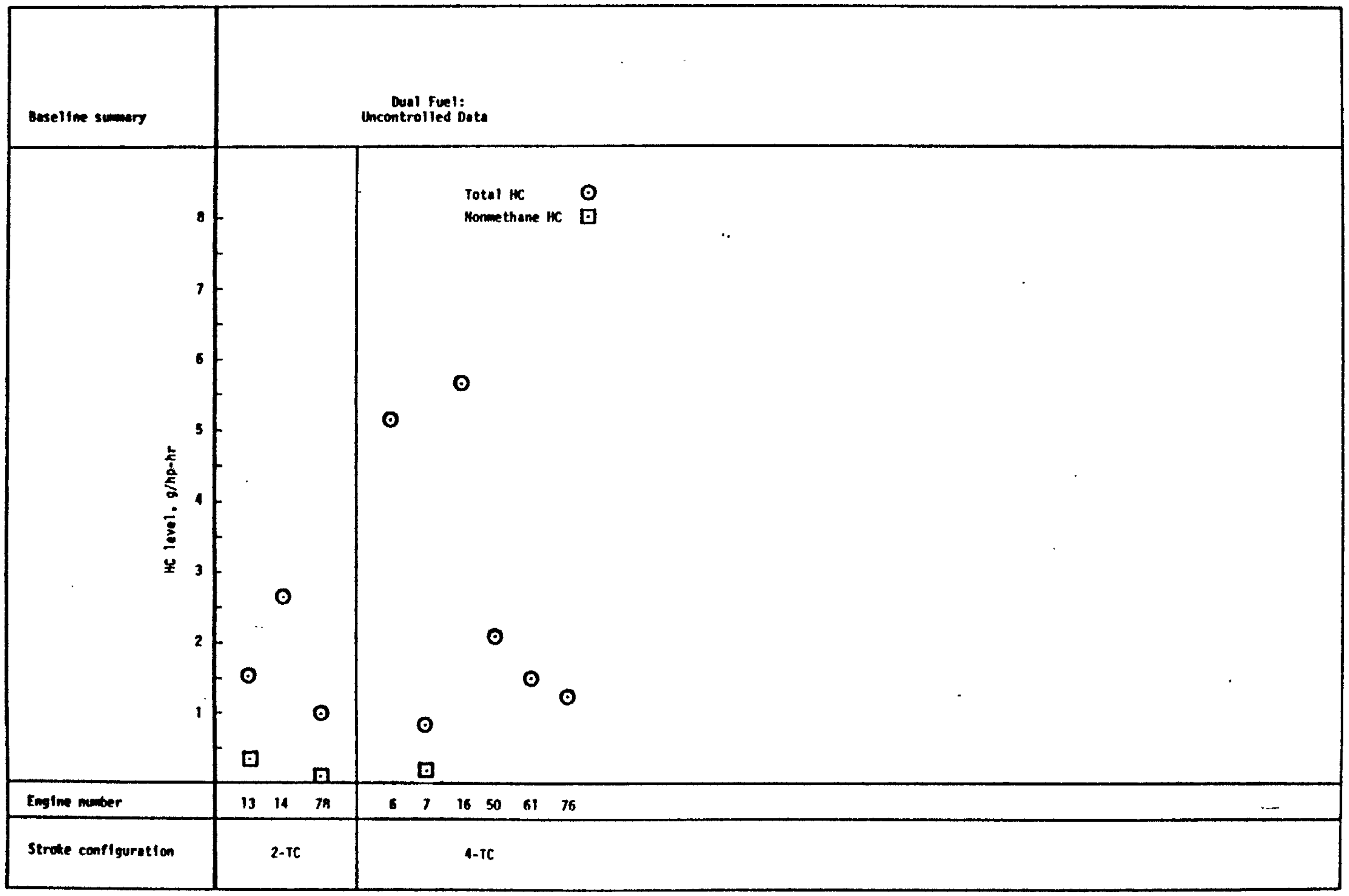


Figure 4-15(b). Uncontrolled HC engines from dual fuel engines.

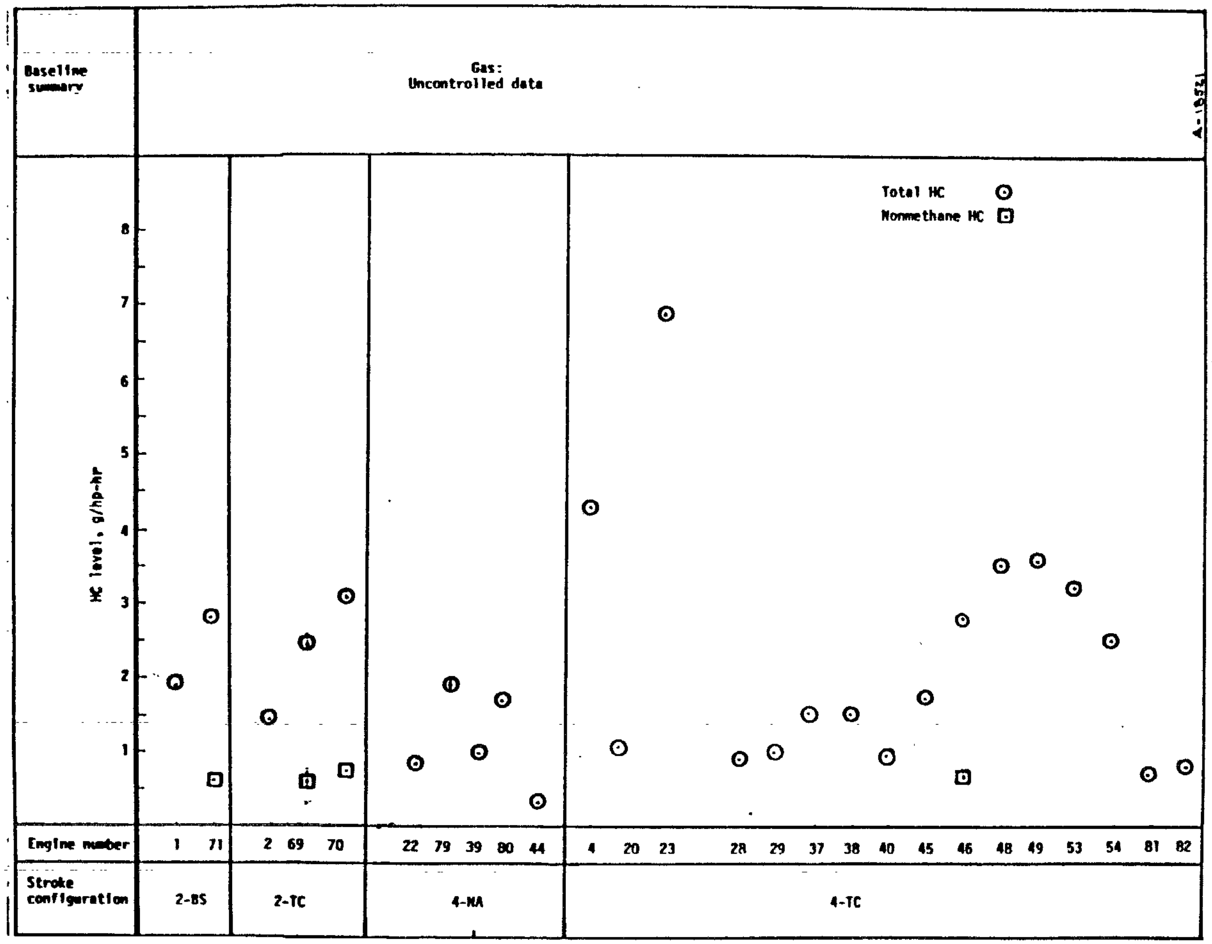


Figure 4-15(c). Uncontrolled HC emissions from gas engines.

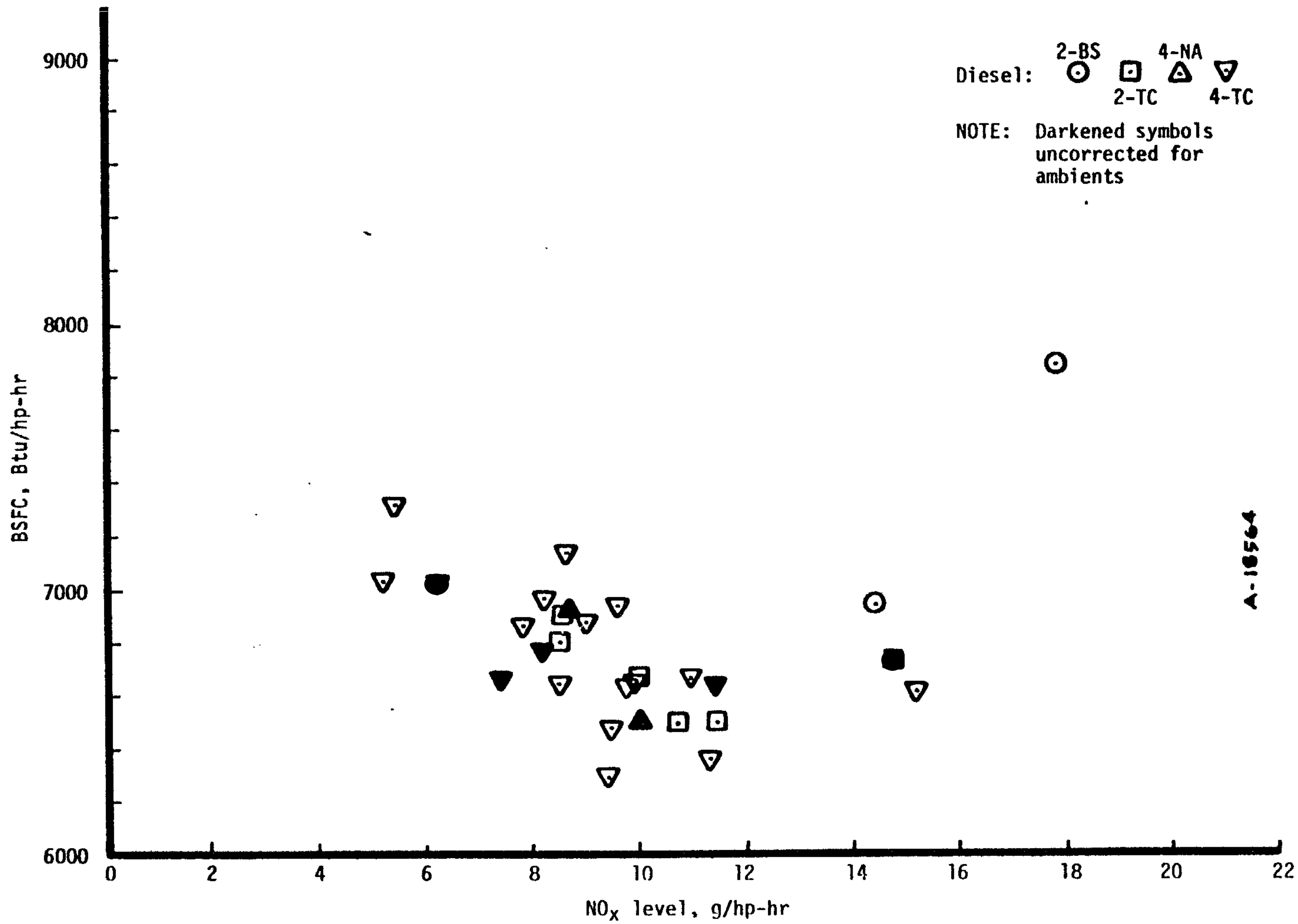


Figure 4-16(a). Uncontrolled NO_x levels versus brake specific fuel consumption (BSFC) for diesel engines.

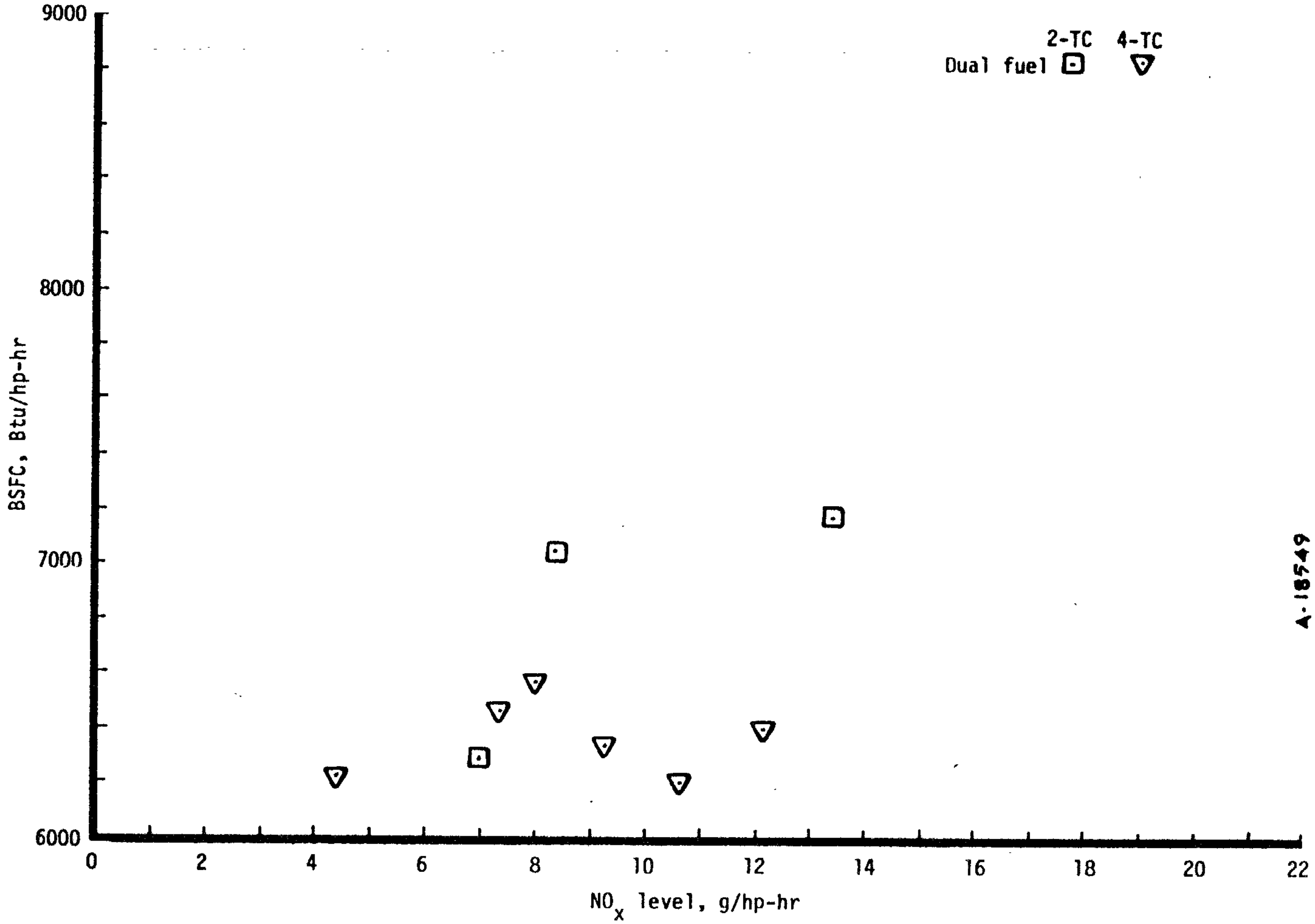


Figure 4-16(b). Uncontrolled NO_x levels versus brake specific fuel consumption (BSFC) for dual fuel engines.

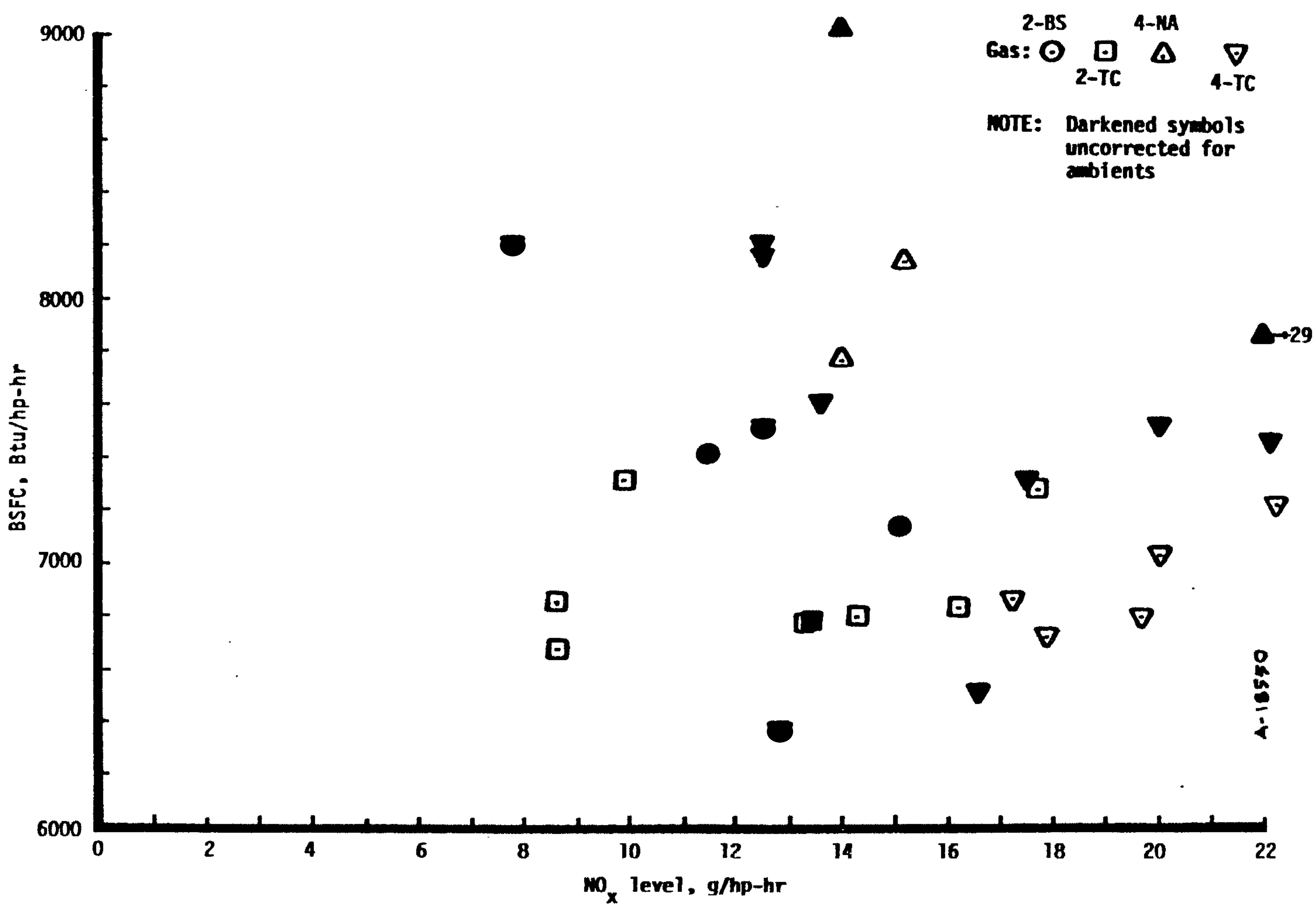


Figure 4-16(c). Uncontrolled NO_x level versus brake specific fuel consumption (BSFC) for gas engines.

important in affecting uncontrolled NO_x emissions and fuel consumption, changes in operating conditions intended to reduce NO_x emissions, generally causing fuel consumption to increase, as discussed in Section 4.4.

As noted in Figure 4-13(a), not all the NO_x data can be corrected for the effect of ambient temperature and humidity. (Ambient correction factors for large-bore engines are summarized in Section 4.2.1.) The effects of ambient variations, differences in measurement practices, and inherent differences in engine design on the variability of uncontrolled NO_x emissions are summarized in Table 4-6[66]. The data samples include only those data from Figure 4-13(a) that could be corrected for ambient variation. As this table indicates, the largest source of variations in data is inherent differences in engine design. This conclusion is similar to that of an investigation of sources of emission data variability in gasoline vehicles[67]. In this study, researchers demonstrated that although measurement and ambient effects were significant, variations among vehicles caused most of the variations in NO_x emissions for a series of tests on similar vehicles. Sources of emissions variability due to engine design are discussed in the following section.

4.3.3 Effect of Engine Variability on NO_x Emissions

The emission data supplied by the manufacturers vary considerably. As discussed above, small variation can be attributed to using different measurements techniques or not correcting for ambient conditions. However, most of the differences in emissions from uncontrolled engines result from: (1) variations in the production of a particular model, (2) variations among different models of the same type (i.e., same strokes/cycle, air charging and fuel), or (3) variations in the number of cylinders for a given model. In

TABLE 4-6. SOURCES OF DATA VARIABILITY FOR UNCONTROLLED NO_X EMISSIONS FROM LARGE-BORE ENGINES

	Mean[a]	Std. Dev.	Sources of Variability[b]		
			Ambient	Measurement	Engine
FUEL	$\overline{NO}_X$, g/hp-hr	$\sigma/\overline{NO}_X$, %	$\Delta A/\overline{NO}_X$, %	$\Delta M/\overline{NO}_X$, %	$\Delta E/\overline{NO}_X$, %
Diesel	10.2	34	6	High = 15 Low = 5	30 to 33
Dual Fuel	9.2	34	8	High = 18 Low = 6	28 to 33
Gas	15.0	25	11	High = 9 Low = 3	20 to 22

[a]Mean values were computed from the corrected data shown in Figures 4-13(a), (b), and (c).

[b]Sources of variability are related according to the law of error propagation (66), i.e.,

$$\sigma = \sqrt{\Delta A^2 + \Delta M^2 + \Delta E^2}$$

This approach assumes each uncertainty is independent of another, but there exists a statistical chance that the uncertainties could occur at the same time.

Ambient corrections are based on factors presented in Section 4.2.1

The estimate for measurement uncertainty includes a high and low value based on information presented in Section 4.2.2

Uncertainties due to engine design (ΔE) were computed from σ, ΔA, and ΔM.

this section, the uncontrolled data base is evaluated for these sources of variability, and furthermore, the data are examined for trends related to differences in engine design, such as speed, torque (bmep), and manifold air temperature. All of these analyses use data corrected for ambient variations by the methods described in Section 4.2.1.

4.3.3.1 Production Variations

It is difficult to quantify variations in emissions among production units of the same model. Up to now, manufacturers have concentrated on obtaining emissions data for different engine models. Since no emission regulations (with the exception of smoke limits) have been in effect for stationary engines, there has been little impetus for manufacturers of large-bore stationary engines to make exhaust measurements of engines leaving the production line. (In general, fewer than 100 units are produced each year for stationary applications by any one manufacturer.) However, Colt and GMC/EMD, as well as numerous manufacturers of smaller bore, heavy duty engines for trucks have reported variations in emissions from production models.

One large volume manufacturer of medium-bore engines has shown that their laboratory units must emit at levels at least 25 percent lower than a performance standard, to insure that their production models will comply with the standard(68). This margin accounts for production variables that effect emission levels in mass produced engines. For 75 percent of this manufacturer's current engines meeting the Federal automotive emission standard of 16 g/hp-hr (NO_x + HC), current variation in 1.34 g/hp-hr (NO_x + HC). Moreover, this manufacturer believes that the magnitude of production variation is independent of the emission level, and this belief is shared by several other manufacturers of medium-bore engines[69]. For large-bore engines, produced individually to higher tolerances, it is anticipated that this variation should be smaller.

Colt reported less than a 3-percent difference between production models in two NO_x measurements of one in 1972, the other in 1975, from a two-stroke, blower-scavenged diesel engine[70]. Such a small difference was unexpected. They suggest that the variation would more likely be of approximately 10 percent for production units, but they have no data to verify this estimate. Colt has measured NO_x levels of production spark ignited engines (2-TC-G) within 3 percent of each other under similar ambient conditions[71].

GMC/EMD has reported average NO_x levels and standard deviations for samples of their 2-TC and 2-BS diesel models[72]. These results are summarized in Table 4-7[73]. As this table suggests, these variations in NO_x levels of production engines may have resulted from ambient variations. Inlet air temperatures varied over a wide range for both the turbocharged and blower-scavenged units, and humidity was not recorded. An attempt was made to determine whether these observed variations in emissions could be due to changing ambient conditions. First, a correction was computed for each extreme of the reported temperature range, using the methodology presented in Section 4.2.1. These two maximum variations were then compared to the reported data, to determine if temperature variations alone could account for the scatter. Next, a correction was computed for both the reported temperature variations and an assumed humidity variation ranging from 35 to 115 grains H_2O/lb dry air. These corrections were then compared with the production variability; the results are listed in Table 4-8.

TABLE 4-7. VARIATIONS IN NO_x EMISSIONS FROM GMC/EMD PRODUCTION ENGINES (Reference 73)

Type	# Cylinders	# Units	Variation (standard deviation as a % of average)	Ambient Temperature
2-TC-D	20	11	± 8	67-97°F
	16	10	± 8	
2-BS-D	16	13	± 6	63-128°F
	12	8	± 5	

[a]Variations are computed for data that are uncorrected for ambient conditions.

TABLE 4-8. POTENTIAL SCATTER IN NO_x EMISSIONS FROM PRODUCTION ENGINES DUE TO AMBIENT HUMIDITY AND TEMPERATURE VARIATIONS

Type	# Cylinders	Inlet Air Temperature Range	Scatter, Uncorrected for Ambient, % of Mean	Correction for Temperature Range[a], % of Mean	Correction for Temperature & Humidity Range[b], % of Mean
2-TC-D	20	67-97°F	± 8	+3 to -2	+16 to -11
	16		± 8	(67 to 97°F)	
2-BS-D	16	63-128°F	± 6	+5 to -9	+15 to -14
	12		± 5	(63 to 128°F)	

[a]Using HD diesel ambient correction factor (see Section 4.2.1.3)

$K_{correction} = 1/(1 + B(T - 85))$ where B = 0.0017 for 2-TC-D (based on 4-TC-D)
= 0.00235 for 2-BS-D
T = test temperature
(Reference temperature of 85°F)

[b]Using HD diesel ambient correction factor

$K_{correction} = 1/(1 + A(H - 75) + B(T - 75))$ where A = -0.00231 for 2-TC-D (based on 4-TC-D)
= -0.00242 for 2-BS-D
H = test humidity ranging from 35 to 115 grains H_2O/lb dry air
(Reference humidity = 75 grains H_2O/lb dry air)

Based on these corrections, temperature variations alone could account for all of the variability in emissions from blower-scavenged engines, but not for turbocharged engines. If humidity were to vary over the range used for the calculations presented in Table 4-8, then differences in ambient temperatures and humidity could account for all the variability reported for these engines.

4.3.3.2 Model Variations

Variations in levels may be attributed to differences in models for a given manufacturer's engines. For example, NO_X levels for a manufacturer's 4-TC-G models may vary due to differences in bore, stroke, turbocharger, configuration (inline cylinders vs. vee), compression ratio, aftercooler, and other engine design parameters. In an effort to identify the magnitude of model-to-model variations, average NO_X levels and standard deviations were evaluated for different models of the same fuel type from each manufacturer.

Table 4-9 presents the results of this study. GMC/EMD, White Alco, and Colt are not included in this table since they each manufacture only one engine model (with different numbers of cylinders) per air charging method. Some of these models are configured for different fuels, for example, Colt markets their 38D8-1/8 opposed piston engine model as a gas, diesel, or dual fuel engine, either blower scavenged or turbocharged. GMC/EMD and White Alco manufacture one basic diesel-fueled, turbocharged design which differs primarily in number of cylinders and speed ratings. GMC/EMD also markets blower-scavenged units.

The other five manufacturers listed in Table 4-9 produce different engine models within a given engine type. NO_X levels reported by Cooper for four 2-TC models varied by an average of 13 percent. These engines differed

TABLE 4-9. VARIATION IN NO_x (g/hp-hr) DUE TO MODEL DIFFERENCES

Manufacturer	Gas[a]								Diesel[b]			
	4-TC				2-TC				4-TC			
	#	AVG	SD	%	#	AVG	SD	%	#	AVG	SD	%
Cooper					4	14.6	1.8	13				
Delaval	2	17.6	0.6	4					3	9.2	0.5	5
Ingersoll-Rand	3[c]	18.0	1.8	10								
White Superior	2[c]	20.4	2.4	12								
Waukesha	3[c]	12.8	0.6	4					2[c]	7.9	0.7	9

[a]Ambient correction for humidity only (see Table 4-2)

[b]Ambient correction for humidity and temperature (see Table 4-2)

[c]No ambient correction, ambients not recorded

SD — Standard deviation in g/hp-hr

% — SD/AVG

— Number of models of same engine type

in bore, speed, number of cylinders, and torque (bmep), but were all operated at the same inlet and manifold air temperatures. Delaval's data indicated only a 4- to 5-percent variation between models for both gas- and diesel-fueled engines. The percent variations shown for Waukesha, Ingersoll-Rand, and White Superior, which were uncorrected for ambient conditions, should not be compared to the Cooper and Delaval results because differences due to ambient conditions could not be factored out. To the extent that conclusions can be drawn from such a small sample size, it appears that NO_x emissions for any type of engine (given strokes/cycle, fuel, and air charging) vary more from manufacturer to manufacturer than among models in a manufacturer's line. Emission variations due to differences among manufacturers could be related to differences in speed, bmep, or manifold air temperature. This possibility is addressed in Section 4.3.3.4.

4.3.3.3 Variations With Number of Cylinders

Several manufacturers have suggested that NO_x levels will vary for a basic engine design depending on the number of cylinders, since the manifold interacts with the turbocharger. Figure 4-17, which is a plot of NO_x level (corrected for ambients) vs. number of cylinders, shows that NO_x levels for 4-TC gas engines decrease significantly as number of cylinders increase, but NO_x emissions from 4-TC diesel and dual-fuel engines do not indicate a trend with number of cylinders.

Figure 4-18 presents a different interpretation. The NO_x levels (corrected for ambients) have been plotted vs. the number of cylinders for each manufacturer's engines, which tends to reduce other sources of emissions variation (such as design differences among manufacturers) that may have been

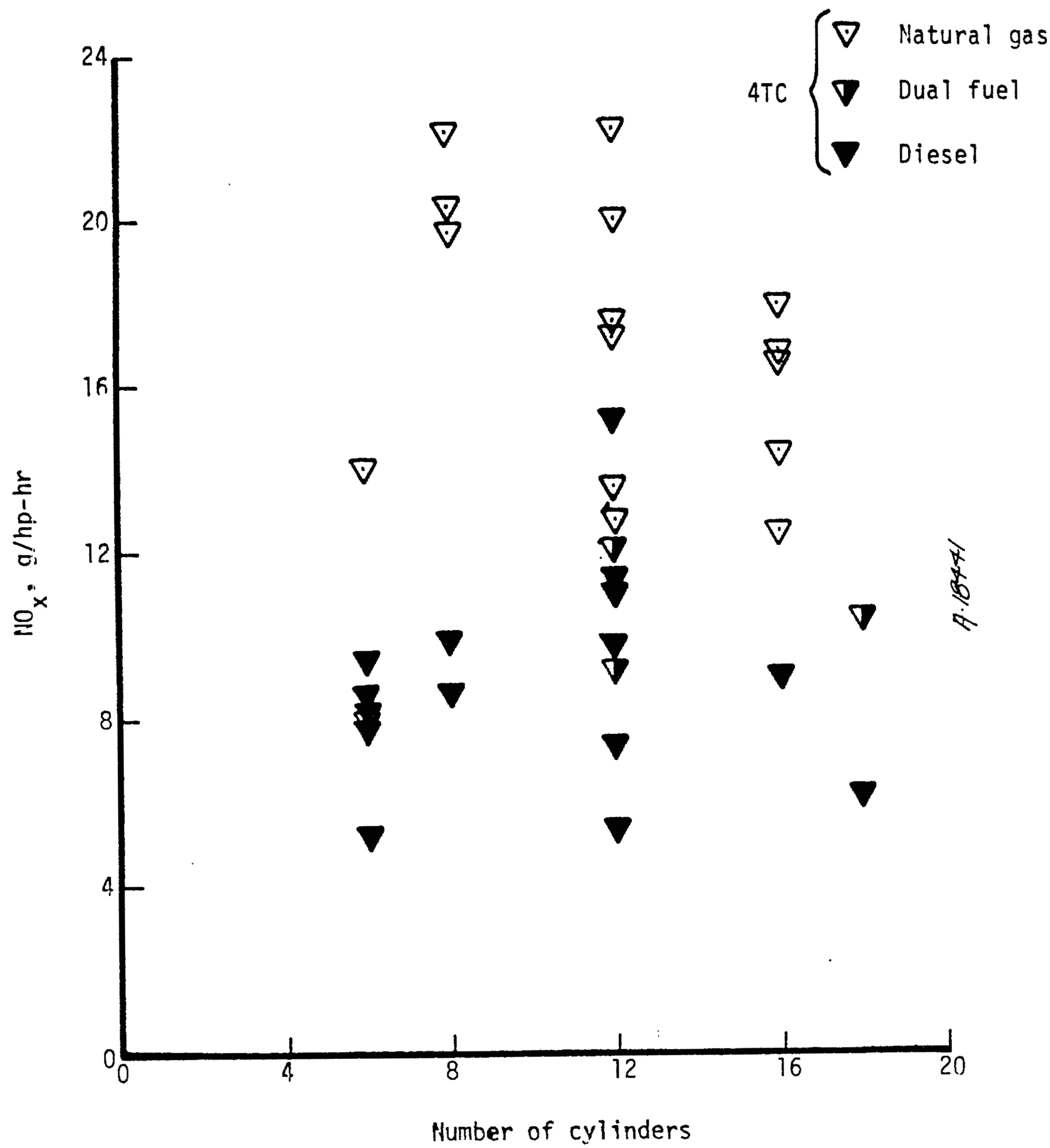


Figure 4-17. NO_x production variation with number of cylinders.

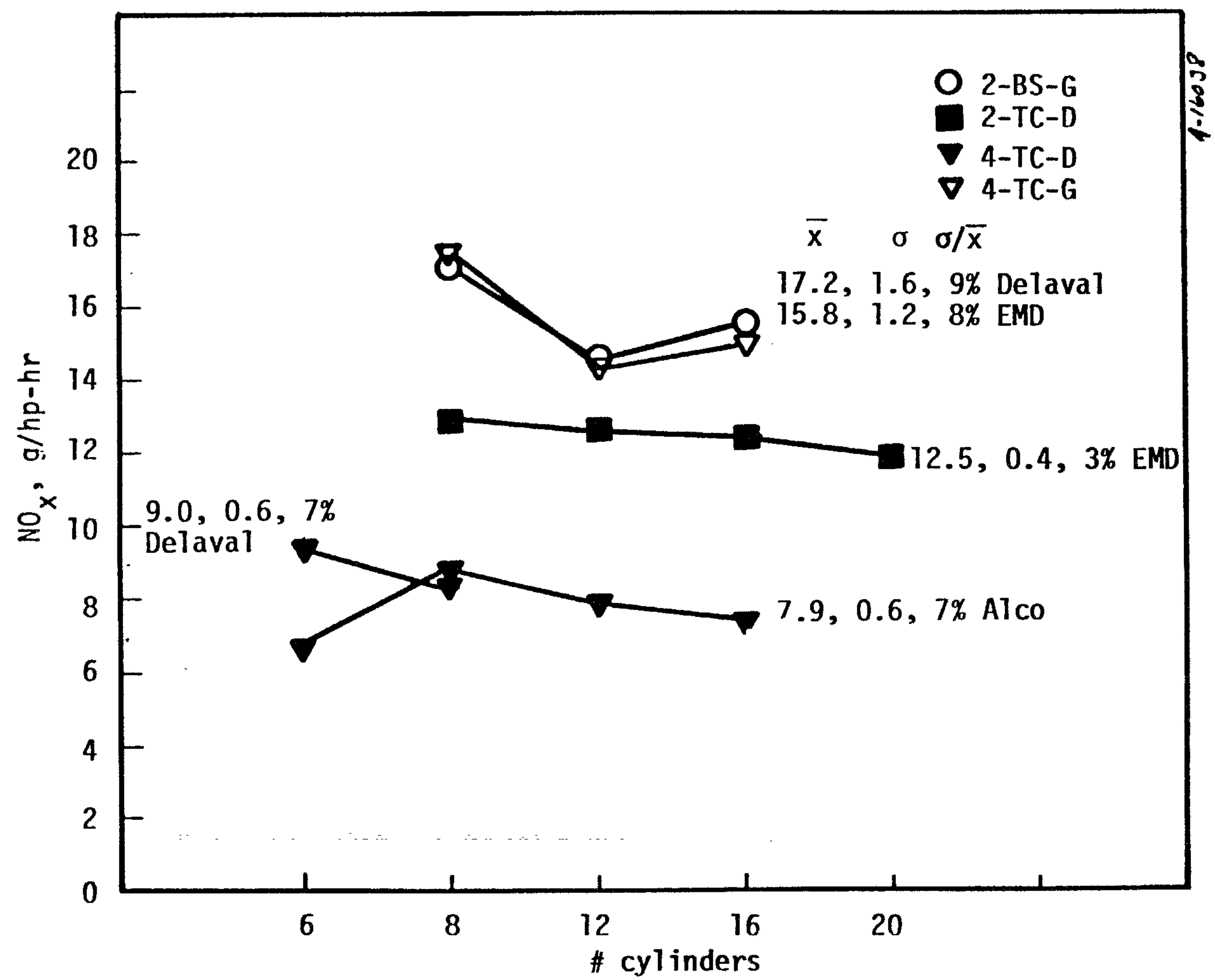


Figure 4-18. Variation in NO_x level with number of cylinders for individual engine models of different manufacturers.

reflected in Figure 4-17. Figure 4-18 indicates that there is no clear trend between NO_x emissions and number of cylinders for either diesel or gas units. The effect of changing the number of cylinders causes uncontrolled NO_x levels to vary from 3 to 9 percent. Because EMD data could not be corrected for ambients, this data may not represent the effect of differences in the number of cylinders.

4.3.3.4 Variations in NO_x Level Due to Other Engine Variables

The results described above suggest that the variations in NO_x levels reported for engines of a given type are probably due to design parameters that differentiate one manufacturer's engines from those of the others. Consequently, uncontrolled NO_x data (corrected for ambients) were plotted vs. speed (rpm), manifold air temperature, and torque (bmep) to reveal any emission trends with these design parameters.

Figure 4-19 illustrates NO_x level variation with speed for two engine types. The data for gas engines indicate increased NO_x emission with in creased speed. This is in contradiction to what one would expect from the reasoning that decreased residence time (increased speed) should result in lowered NO_x emissions. Apparently other factors (e.g., increased cylinder temperature or inherent design differences among different engines) are responsible for this trend. The 4-TC dual fuel and diesel NO_x levels appear to decrease somewhat with increasing speed, as would be expected for lower exhaust gas chamber residence times.

Variations in NO_x level with manifold air temperature are shown in Figure 4-20. The 4-TC-G NO_x levels appear to be very sensitive to the design air manifold temperature, but the diesel and dual fuels NO_x levels do not.

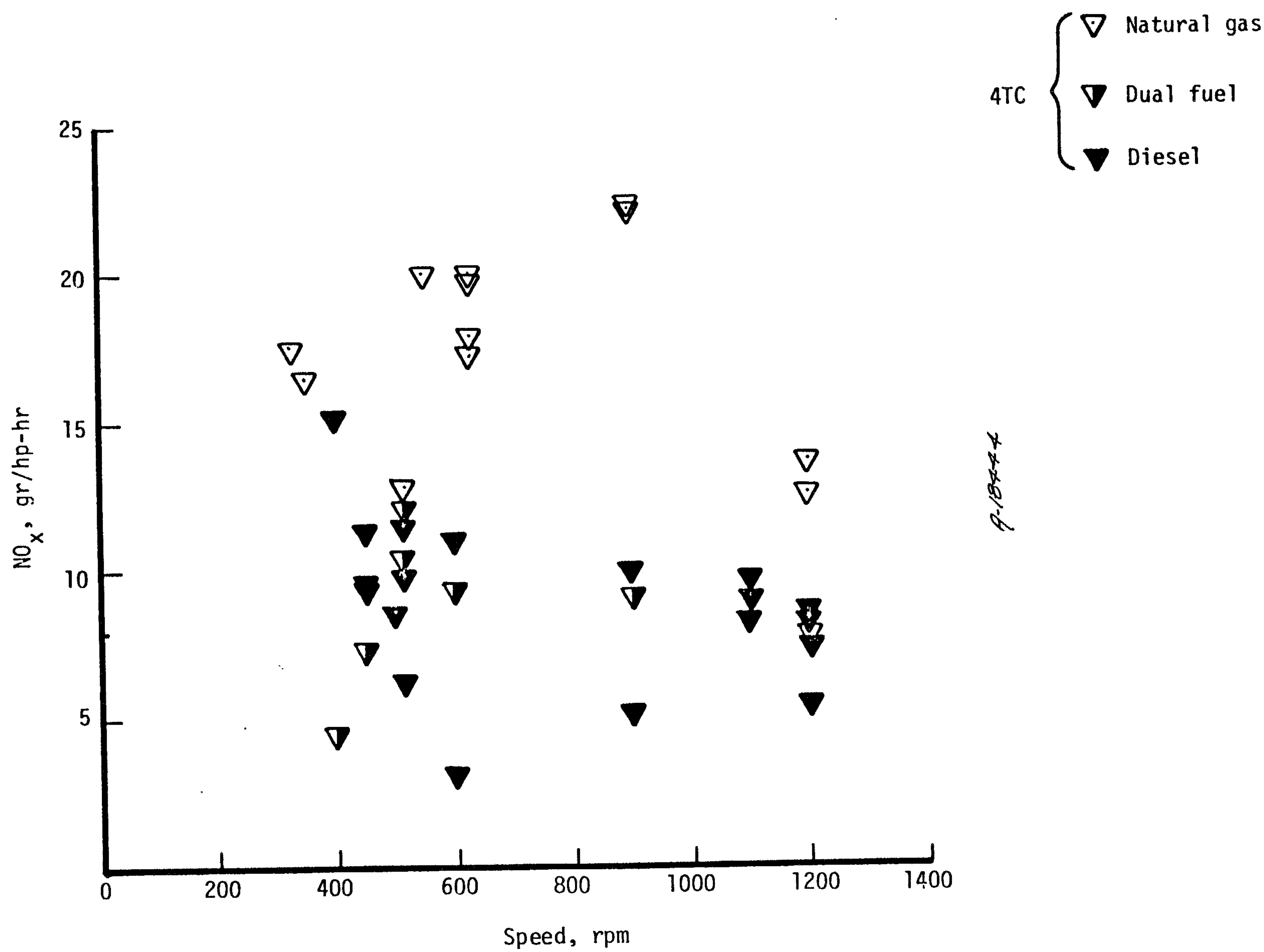


Figure 4-19. Variation in NO_x level with speed.

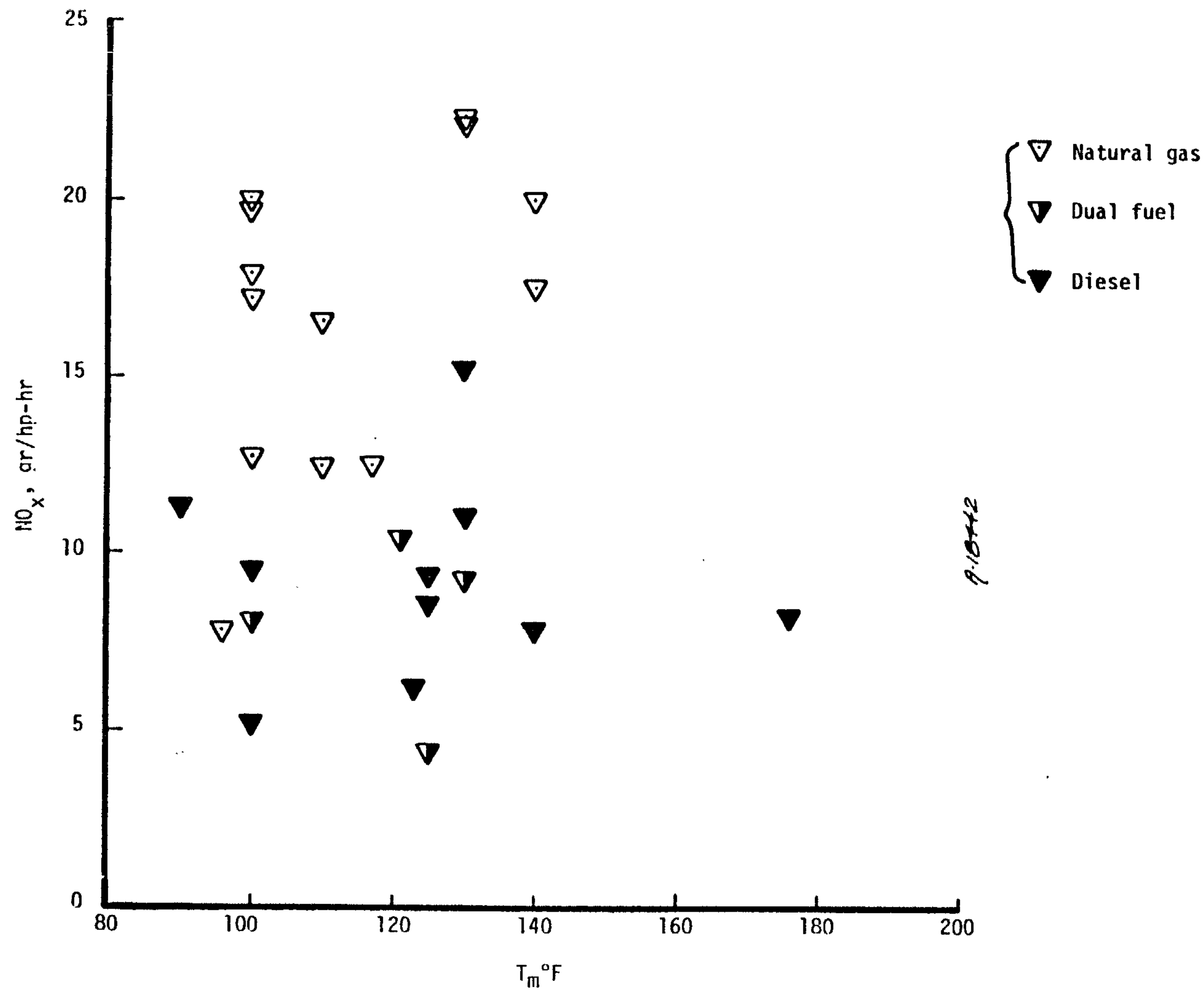


Figure 4-20. NO_X variation with manifold temperature.

These results confirm expectations that NO_x levels from premixed, vaporized fuel combustion in SI engines may be strongly influenced by the degree of aftercooling. On the other hand, CI engines are characterized by droplet combustion, and NO_x production under these conditions would depend more on local A/F ratio than on overall air temperature.

Finally, Figure 4-21 illustrates the wide variation of NO_x level with torque (bmep) for 4-TC-G and 4-TC-D, DF units. If the cluster of CI data around 150 psi is ignored, a trend of decreasing brake-specific NO_x emissions with increasing bmep for these units is apparent. No trend is apparent for SI units, except that they are generally not manufactured with bmep's exceeding 200 psi.

Based on these preliminary studies, it appears that certain engine design parameters may explain more of the variation NO_x levels for engines of a given type than variations in ambient humidity or temperature. That is, NO_x emissions for any type engine (given strokes/cycle, fuel, and air charging) vary more from manufacturer to manufacturer than they do among models within a manufacturer's line. Differences among manufacturers are related to differences in speed, torque (BMEP), manifold air temperature, and combustion chamber design. For example, limited data show that NO_x emissions from 4-TC diesel and dual fuel engines decrease as speed increases. NO_x emissions from 4-TC, natural gas (SI) engines increase directly as design manifold air temperature increases. However, no clear trends can be established for the effects of number of cylinders and torque (bmep). These observations suggest that some form of weighted average is required to characterize uncontrolled NO_x emissions from each of the three fuels. This approach is discussed in the following sections.

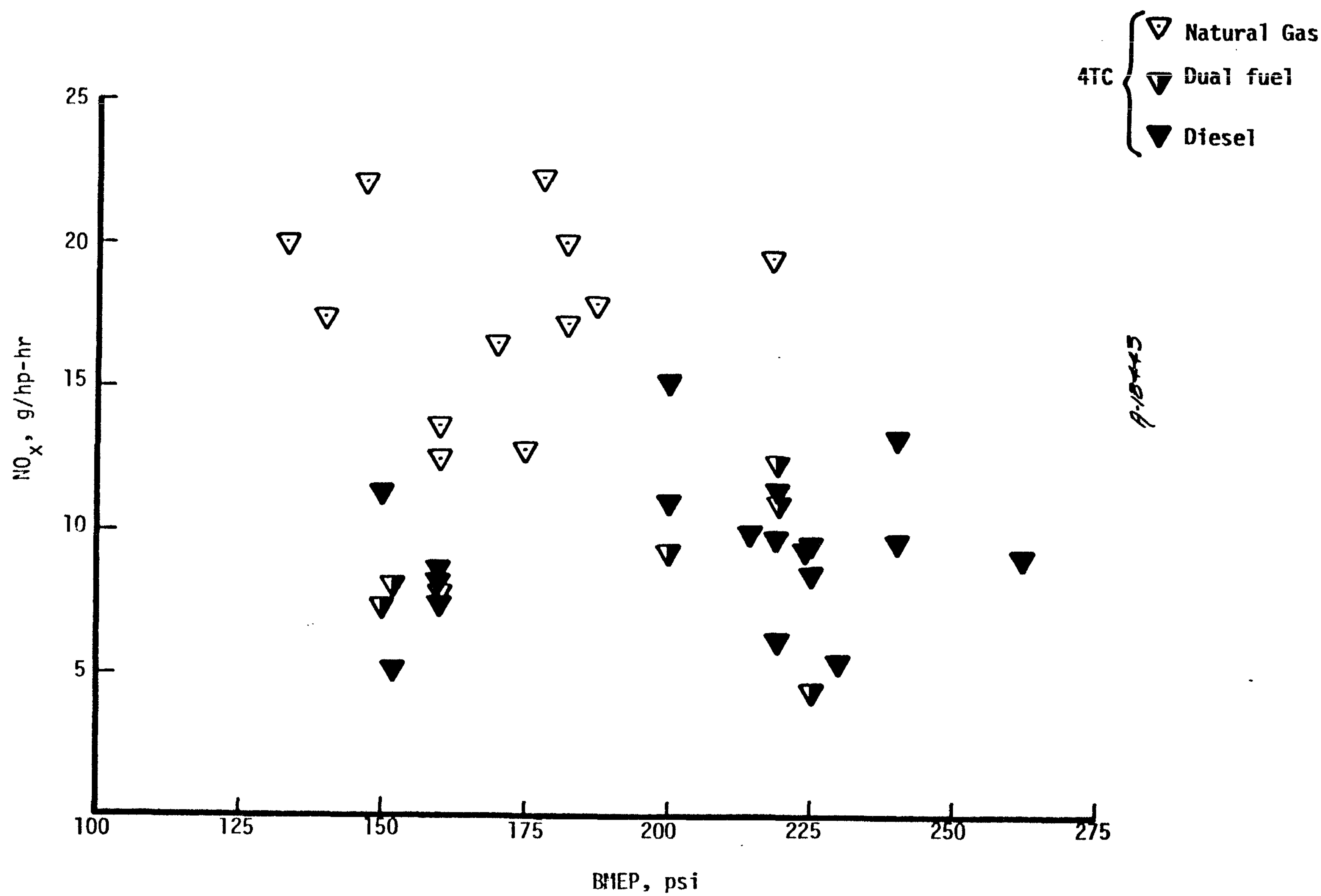


Figure 4-21. NO_x variation with torque (bmep).

4.3.4 Sales-Weighted Uncontrolled Emissions

Since the sources of variability due to engine design cannot be specifically identified, a procedure is required to characterize uncontrolled emission levels of engines which are sold for similar applications.

The procedure adopted here is to compute a weighted, average uncontrolled emission level for engines in the diesel, dual fuel, or natural gas categories. The three weighted levels are based on sales of engine horsepower during the past 5 years for domestic applications. Sales of horsepower to standby services were excluded from this computation, since engines sold for standby applications will be exempted from standards of performance (see Chapter 9). Therefore, these engines should not influence the selection of regulated emission levels.

The sales-weighted averages for diesel, dual fuel, and natural gas engines are presented in Figure 4-22, which also show each manufacturer's uncontrolled NO_x data. The weighted averages are based on data corrected for ambient conditions where possible. The weighted average uncontrolled NO_x level for diesel engines is 11.0 g/hp-hr, for dual fuel units, 8.1 g/hphr, and for natural gas engines, 15.0 g/hp-hr. The emission reductions discussed in Section 4.4 and summarized in Chapter 6 can then be applied to these levels to determine potential regulated levels of NO_x.

Measurement uncertainties are associated with each of these weighted levels and are shown in Table 4-10. These uncertainties should be applied to the controlled NO_x levels that are determined by applying the NO_x reductions demonstrated by the alternative control systems.

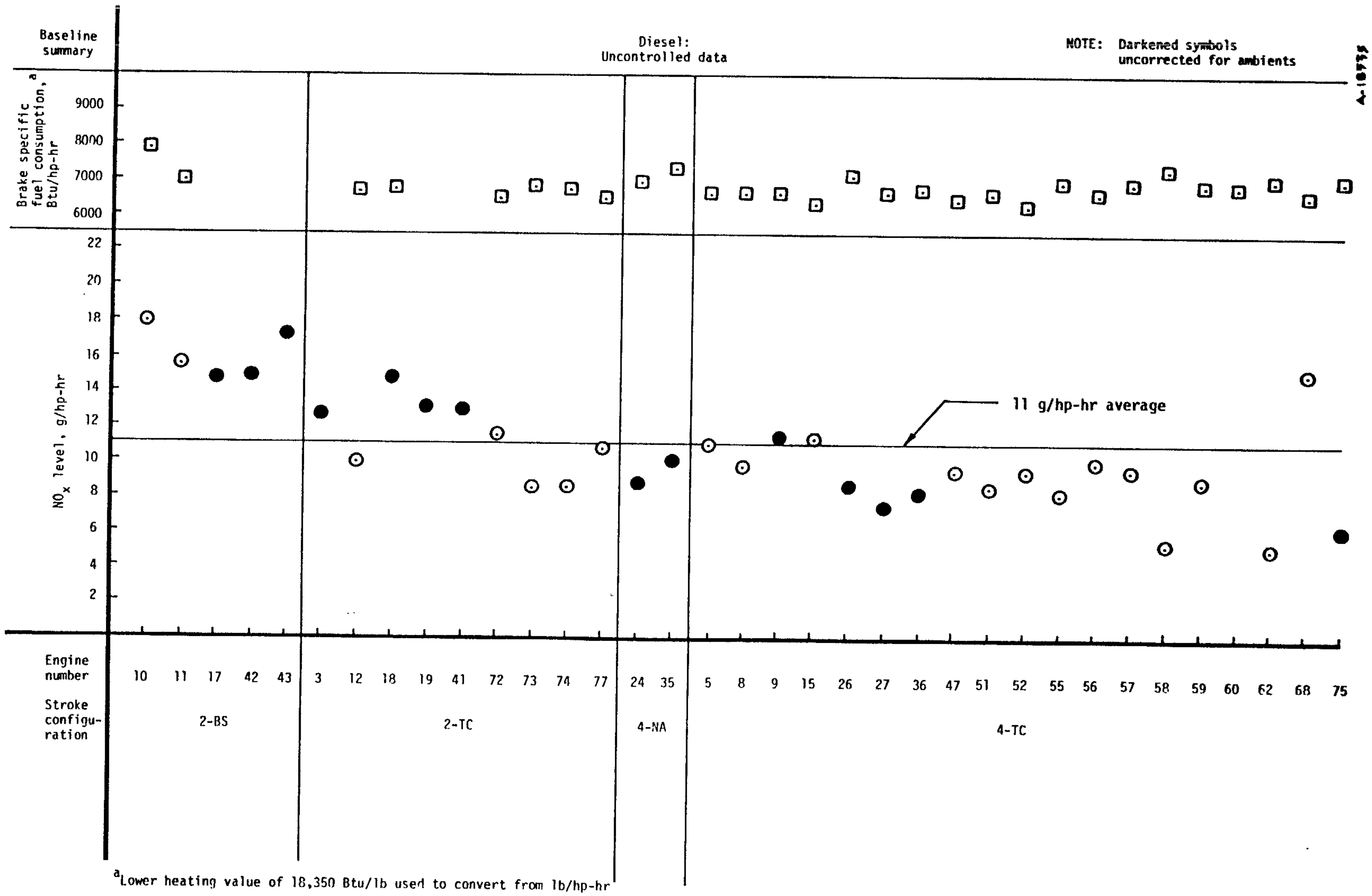


Figure 4-22(a). Sales-weighted uncontrolled NO_x emissions for diesel engines.

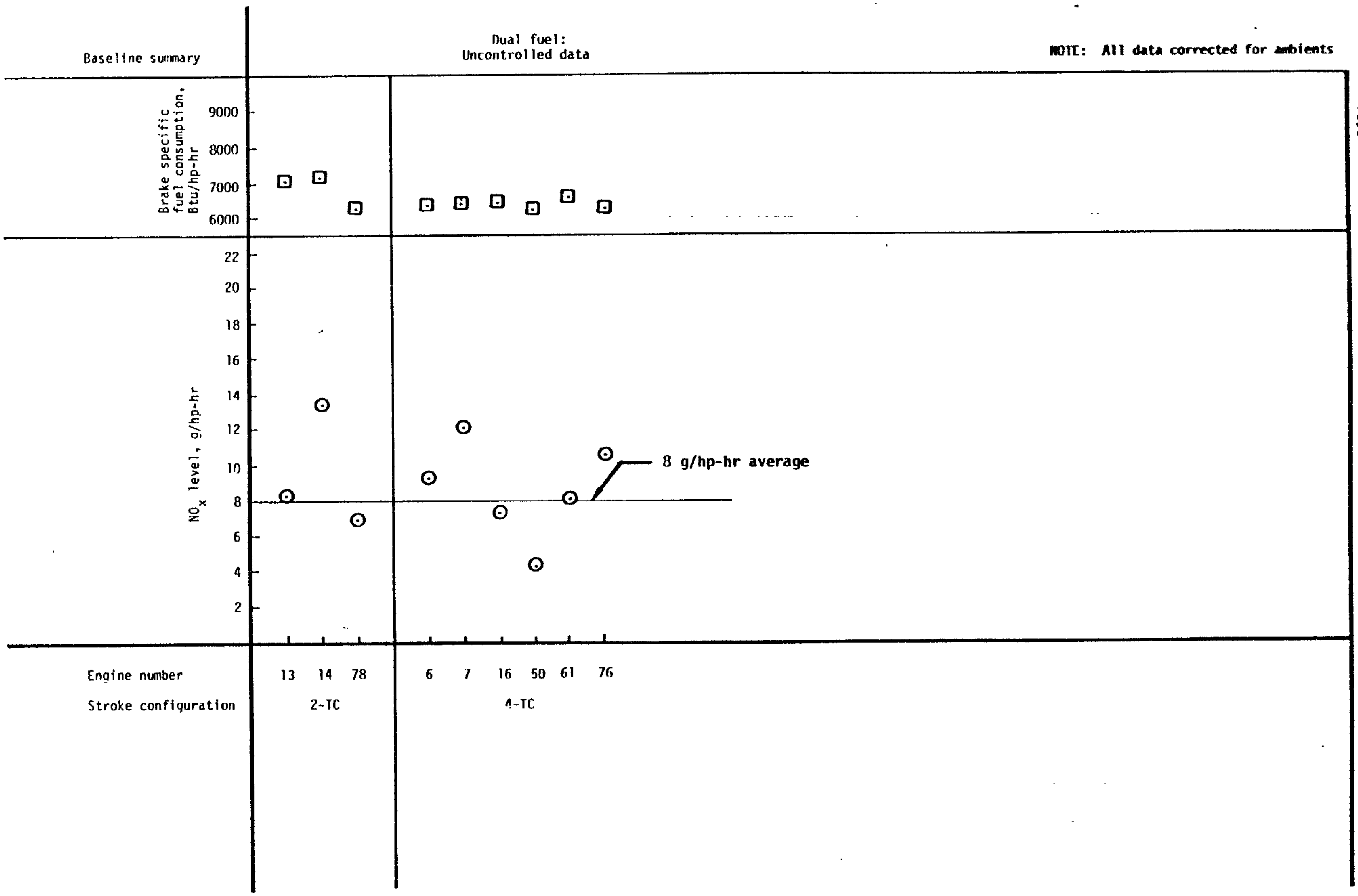


Figure 4-22(b). Sales-weighted uncontrolled NO_x emissions for dual fuel engines.

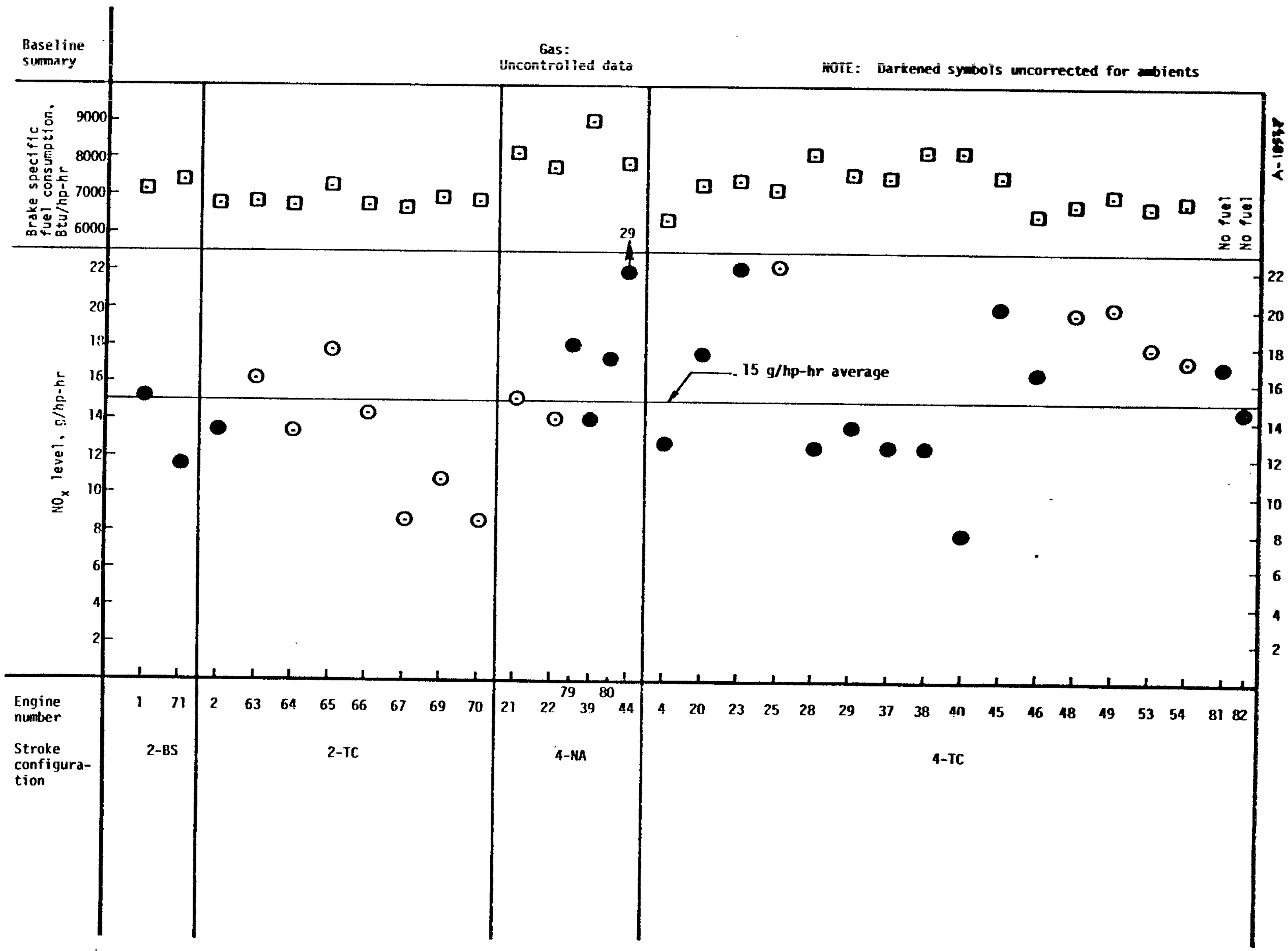


Figure 4-22(c). Sales-weighted uncontrolled NO_x emissions for gas engines.

TABLE 4-10. WEIGHTED MEASUREMENT UNCERTAINTIES FOR SALES WEIGHTED NO_x LEVELS

Fuel	Upper, g/hp-hr	Lower, g/hp-hr
Diesel	1.0	0.9
Dual Fuel	1.0	0.4
Natural Gas	1.3	0.5

4.4 NO_x EMISSION REDUCTION TECHNIQUES

This section describes techniques that have been used, or are being evaluated for use, to control NO_x emissions from IC engines. Sections 4.5 and 4.6 discuss control techniques that are designed primarily to reduce other pollutants (HC, CO, smoke). Since SO_x emissions are directly related to the fuel sulfur content, these emissions are discussed in 4.4.13, combustion of nonstandard fuels.

The data presented here come from tests on engines whose operating conditions were altered or which were equipped with emission reduction devices. These tests were conducted in manufacturers' laboratories rather than in field installations. The discussion of each potential control technique centers on how the technique works, its effectiveness, resulting fuel penalties, effects on other pollutant emissions, technical limitations to its applications, and cost implications (i.e., additional fuel, maintenance, or hardware expense incurred by the application of the control).

Most techniques for controlling emissions from IC engines involve engine modifications rather than add-on tail gas treatment facilities. Engines are designed for optimum operation within one or more of the following constraints: application, initial cost, fuel consumption, maintenance requirements, reliability, and commitment of a company's engineering staff to a design.[5/] Each engine design satisfies the constraint

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[5/] Stationary reciprocating IC engines, and particularly the large ones, may be required to deliver thousands of hours of continuous operation at rated load under varying ambient conditions without significant maintenance, or to start without failure by remote control and deliver full power within 10 seconds. Since these are severe demands, manufacturers feel committed to a proven design and are, therefore, reluctant to make significant design changes (e.g., changed piston or cylinder shape or strokes per cycle).

differently. For example, one engine may operate at 4° BTDC while another at 5° BTDC to meet the same NO_x emission level. Therefore, the data are grouped by engine type and fuel in the tables and graphs that follow. In addition, whenever there is a specific, known reason why one type of engine responds differently to the application of controls than does another, these differences are explained in the accompanying discussion of the control technique.

The reductions in NO_x shown here were achieved by investigators for current production engines. In general, no attempt was made to optimize the engine for the controlled settings (i.e., decrease fuel consumption, reduce maintenance, etc.). Thus, these results must be viewed as those achievable if no attempt is made to reoptimize an engine's controlled setting.

As discussed in Section 4.2.2, the manufacturers' data were measured using one of four measurement practices (EPA, DEMA, SAE, EMD). Although differences in three of these practices relative to EPA's may cause uncertainties in the reported levels, the data are considered adequate for the purpose of setting standards of performance since these are small in comparison to those in emissions due to inherent differences in engine design. (Measurement uncertainties for uncontrolled emissions are discussed in Sections 4.3.2 and 4.3.4.) Furthermore, the reported emissions data have been corrected to standard conditions of humidity and temperature (when ambient data were recorded) using the ambient correction factors presented in Section 4.2.1 Dashed lines on the figures in this section indicate NO_x reductions after ambient correction.

The control systems discussed in this section are listed below in their order of presentation.

1. Derating (D)
2. Retard (R)

3. Changed air-to-fuel ratio (A/F)
4. Turbocharging with aftercooler (TC)
5. Reduced manifold air temperature (MAT or M)
6. Exhaust gas recirculation (EGR) -- internal (IE) and external (EE)
7. Water induction (H_2O)
8. Combustion chamber redesign (CCR)
9. Catalytic converters
10. Combinations of the above

Several abbreviations will be used on the charts and tables in this chapter; they are listed in Table 4-11. Fuel consumption data on the charts and tables are based on a lower heating value (LHV) of 18320 Btu/lb (10160 kcal/kg) for No. 2 diesel oil.

A qualitative summary, by pollutant, of the effect of each control technique on each engine type as shown by the available data is presented in Table 4-12. Sections 4.4.1 through 4.4.10 give quantitative results for each of these techniques. Graphs are presented that show (1) the NO_x reduction for the largest degree of control applied for each manufacturer and (2) the effect on emissions and fuel consumption as the amount of control is varied. The information presented in the second set of graphs has been normalized by the baseline or uncontrolled level. This condition is denoted on the graphs with a subscript "U" for uncontrolled. The controlled condition is denoted with a subscript "C" for the controlled level. Section 4.4.11 summarizes the data presented for each of the above control approaches.

Then in Section 4.4.12, the effect of NO_x control on the emission of other pollutants is examined. This review will help to illustrate whether standards of performance may be required for other pollutants in addition to

TABLE 4-11. ABBREVIATIONS FOR ENGINE TYPE AND EMISSION CONTROL TECHNOLOGY

Abbreviation	Explanation
Fuel	
D	Diesel
DF	Dual Fuel
G	Gas (i.e., natural gas)
Strokes/Cycle	
2	2-stroke/cycle engine
4	4-stroke/cycle engine
Air Charging	
BS	Blower scavenged
NA	Naturally aspirated
TC	Turbocharged (and intercooled)
Control Technology[a]	
D	Derating
R	Retard
TC	Turbocharged (and intercooled)
A or A/F	Increased air-to-fuel ratio
M or MAT	Decreased inlet manifold air temperature
EGR(I)	Exhaust gas recirculation — internal
EGR(E)	Exhaust gas recirculation — external
S[b]	Increased speed
INJ	Modified injectors
H_2O	Water induction
W/F or w/f	Water-to-fuel mass induction ratio
PC	Precombustion chamber
VT	Variable throat precombustion chamber
CCR	Combustion chamber redesign
Cat	Catalytic converter

[a]RM and RMA are used to denote the combined use of retard, decreased air temperature, and increased air-to-fuel ratio.

[b]Increased speed included because data are available, but it is not considered to be a viable control technique (Subsection 5.3.12)

TABLE 4-12(a). EFFECTS OF CONTROLS ON ENGINES LARGER THAN 350 IN^3/CYL: NO_x EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control \ Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
Derate	↓			↓				↓	↓	↓	↓	↓
Retard	↓	↓		↓		↓		↓	↓	↓	↓	↓
A/F				↓		↓		↓	↓	↓	↓	↓
TC	↓	—		—		—		—	↓	—	↑	—
MAT	↓	↑		↓		↓		↓	↓	↓		↓
INJ	↓	↑↓										
EGR(I)				↑				↑	↓		↓	
EGR(E)		↓				↑						
H_2O	↓			↓				↓			↓	↓
CR			↓	↓								

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 4-12(b). EFFECTS OF CONTROLS ON ENGINES LARGER THAN 350 IN^3/CYL: CO EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control \ Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
Derate	↑			↓				↑↓	↑	↑	↑	↑
Retard	↑	↑		↑		↑		↑	↑	↑		↓
A/F				↑↓		↑		↓		↑	—	
TC	↑↓								↓		↓	
MAT	↓	↓		↑↓		↑		↑↓	↑	↑		↑
INJ	↓	↑↓										
EGR(I)				↓				↓	↑		—	
EGR(E)		↑				↑						
H_2O	↓			↑↓				↓			↑	↑
CR												

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

TABLE 4-12(c). EFFECTS OF CONTROLS ON ENGINES LARGER THAN 350 IN^3/CYL: HC EMISSIONS

Fuel	Diesel				Dual Fuel				Natural Gas			
Strokes/Cycle	Two		Four		Two		Four		Two		Four	
Control / Air Charging	BS	TC	NA	TC	BS	TC	NA	TC	BS	TC	NA	TC
Derate				↑				↑	↑	↑	↑	↑
Retard	—	↑↓		↑↓		↑		↓	↑	↑	↑	↑
A/F				↑↓		↑		↑		↑		↑
TC	↓								↓			
MAT	↑	↑↓		↑↓		↑		↓↑	↑	↑		↑
INJ	↓	↓	↓									
EGR(I)				↓				↑	↑			
EGR(E)		—				↓						
H_2O	↑			↑↓				↑			↑	↑
CR												

↑ Denotes emission increase with application of control
↓ Denotes emission decrease with application of control
↑↓ Denotes conflicting data with application of control
— Denotes no change in emissions with application of control
Blank indicates no data available on effect

NO_x. Finally, Section 4.4.13 discusses the effect of burning nonstandard fuels on emissions from stationary IC engines.

The data that appear in this chapter can also be found in Appendix C.1, where they are tabulated by engine. This appendix presents all the available data on emissions and fuel consumption for large-bore engines.

4.4.1 Derating

When a manufacturer advertises or sells an engine, he guarantees that it will produce a given power at a stated speed. These conditions are called "rated conditions" and can be specified either for maximum, intermittent, or continuous operating conditions. The maximum rating usually refers to the peak power that can be achieved by the engine, but manufacturers generally recommend that the engine not be operated at this level. Intermittent ratings typically indicate the power output that the engine can produce for a 1-hour period with at least a 1-hour period of operation at, or below, the continuous rating before the next surge to the intermittent level[74]. Continuous rating, of course, applies to uninterrupted operation (e.g., 24 hours per day, 365 days per year with shutdowns for maintenance only).

An engine can be derated by restricting its operation to a lower level of power production than normal for the given application. The effect of derating is to reduce cylinder pressures and temperatures and thus to lower NO_x formation rates. Although NO_x exhaust concentrations (i.e., moles of NO_x per mole of exhaust) are reduced, it is quite possible for this reduction to be no greater than the power decrease. In such a case brake specific emissions (i.e., grams NO_x per horsepower-hour) are not reduced. This is especially true for four-stroke turbocharged engines as shown in Figure 4-23[75]. In addition, air to fuel ratios change less with derating for

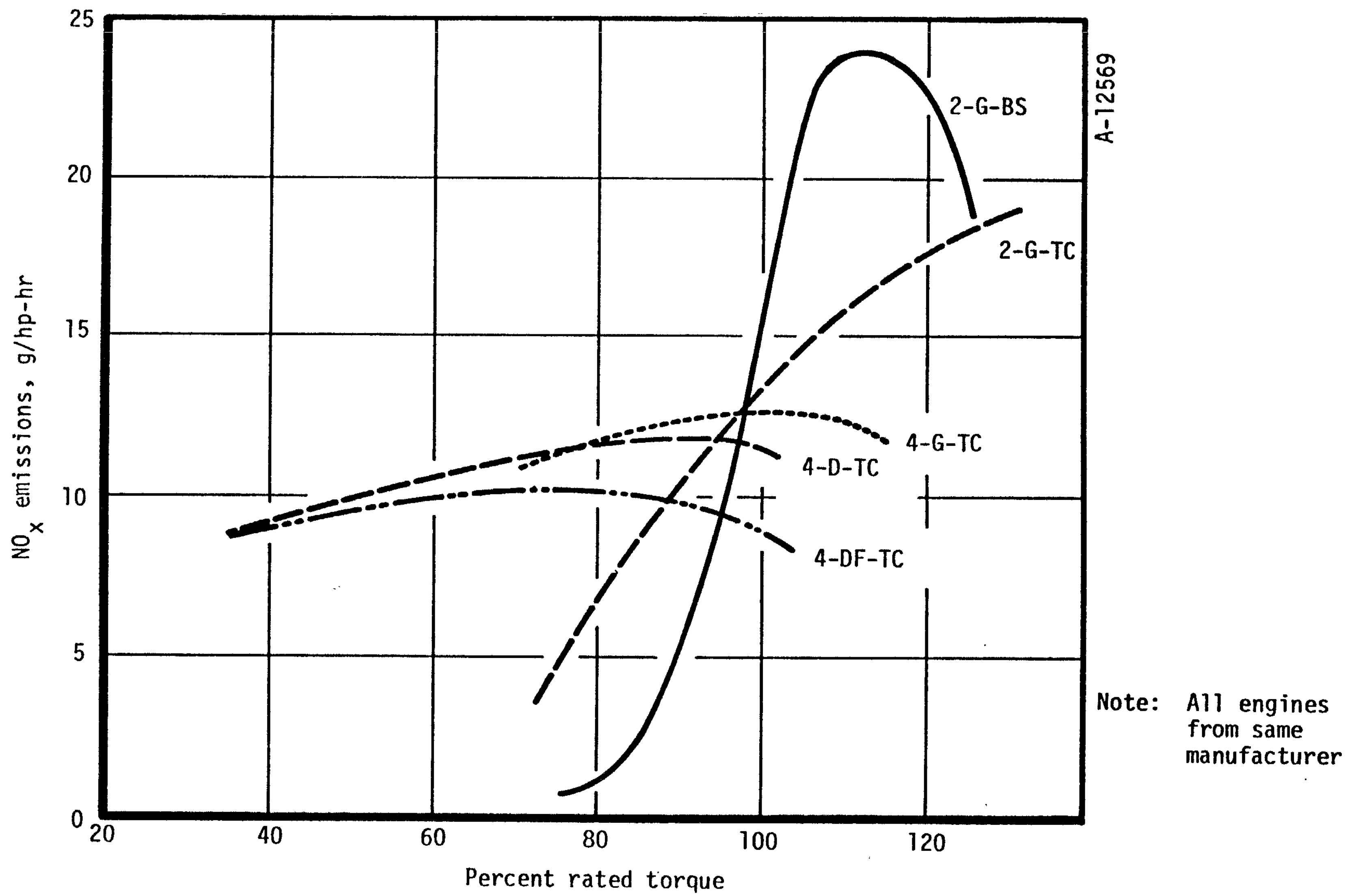
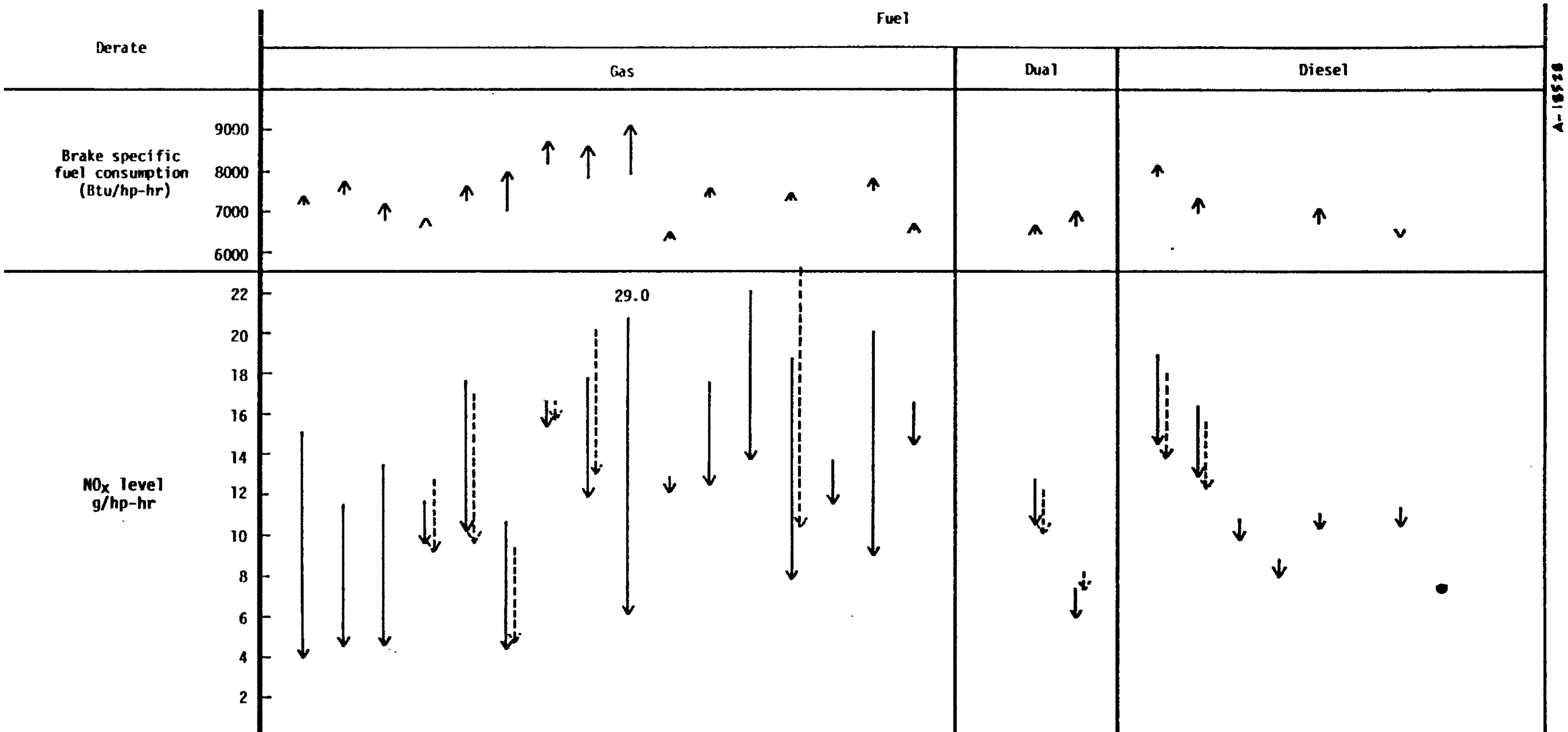


Figure 4-23. NO_x emissions versus torque at constant speed (Reference 75).

turbocharged engines than for naturally aspirated or blower-scavenged units. Thus NO_x emissions are less responsive to derating for turbocharged engines. Derating also reduces the engine's operating temperature, which then results in higher CO and HC emissions. This happens because the temperature dependent reactions that reduce these pollutants are less active(76).

Demonstrated NO_x emission reduction levels due to derating are shown in Figure 4-24 for a number of different engine types and fuel. Based on these data, emission reductions ranged from 1.2 to 23.0 g/hp-hr for naturally aspirated or blower-scavenged engines and from 0.2 to 10.8 g/hp-hr for turbocharged units. Since these results were obtained with varying amounts of derating, it is more informative to compare the effectiveness of this emission control technique on a normalized basis -- i.e., percent NO_x reduction per percent derate. On this basis, results for naturally aspirated or blower-scavenged engines varied from 0.25 to 6.2, whereas those for turbocharged units varied from 0.01 to 2.6. No relationship was found between normalized effectiveness and uncontrolled emission level, number of strokes per cycle, or fuel.

Figure 4-25 illustrates the effect of different amounts of derating on NO_x emissions and fuel consumption for diesel, dual fuel, and gas engines, respectively. Figure 4-25(a) shows that derating decreases brake specific NO_x emission from some diesel engines, but increases them from others. In general, NO_x reductions range from 2 to 25 percent for 25 percent derating and brake specific fuel consumption increases range from 2 to 5 percent. With 50-percent derating, NO_x reductions range from 15 to 45 percent and fuel penalties from 4 to 16 percent. Engines No. 10 and 11, both two-stroke blower-scavenged, achieved the largest NO_x reduction (and highest fuel penalties).



Fuel	Gas																Dual		Diesel						
Percent derate	12	20	25	13	20	25	25	25	25	14	25	25	25	25	25	25	18	25	25	·5	25	25	38	27	25
Normalized reduction[a]	6.2	3.0	2.6	1.4	2.1	2.3	0.3	1.3	0.4	0.3	1.1	1.5	2.3	0.6	2.2	0.5	0.9	0.8	0.9	4.2	0.3	0.5	0.2	0.3	0.2
Uncontrolled fuel consumption	7127	7400	6775	6760	7265	6942	8130	7760	7840	6374	7300	7440	7220	7590	7500	6500	6400	6570	7844	6951	7175	6930	6677	6366	6667
Percent increase	2.6	4.1	6.1	0.3	4.3	14.4	6.4	9.6	15.3	0.8	3.0	1.7	2.8	--	3.5	2.0	3.1	6.1	2.6	4.1	--	-5.5	4.8	-0.7	--
Air charging	BS	BS	TC	TC	TC	TC	NA	NA	NA	TC	TC	TC	TC	TC	TC	TC	TC	TC	BS	BS	TC	NA	TC	TC	TC
Strokes/cycle	2	2	2	2	2	2	4	4	4	4	4	4	4	4	4	4	4	4	2	2	2	4	4	4	4
Engine number	1	71	2	64	65	69	21	22	44	4	20	23*	25	29	45	46	7	61	10	11	12	24*	5	15	27

[a]Percent NO_x reduction/percent derate.

[b]Speed reduction, const. torque.

Figure 4-24. Effect of derate on NO_x emissions.

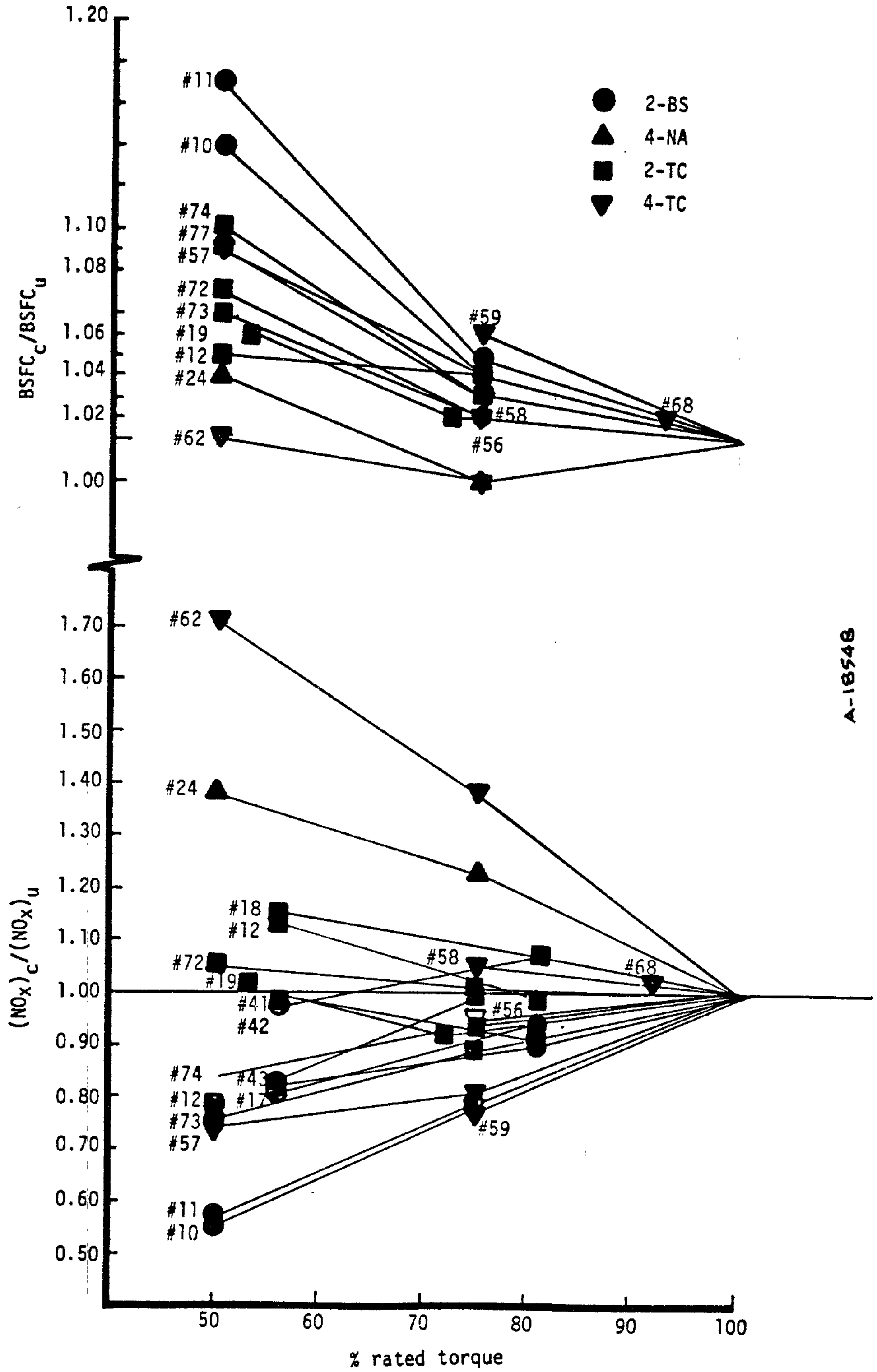


Figure 4-25(a). Effect of different amounts of derate on NO_X emissions and fuel consumption from diesel engines.

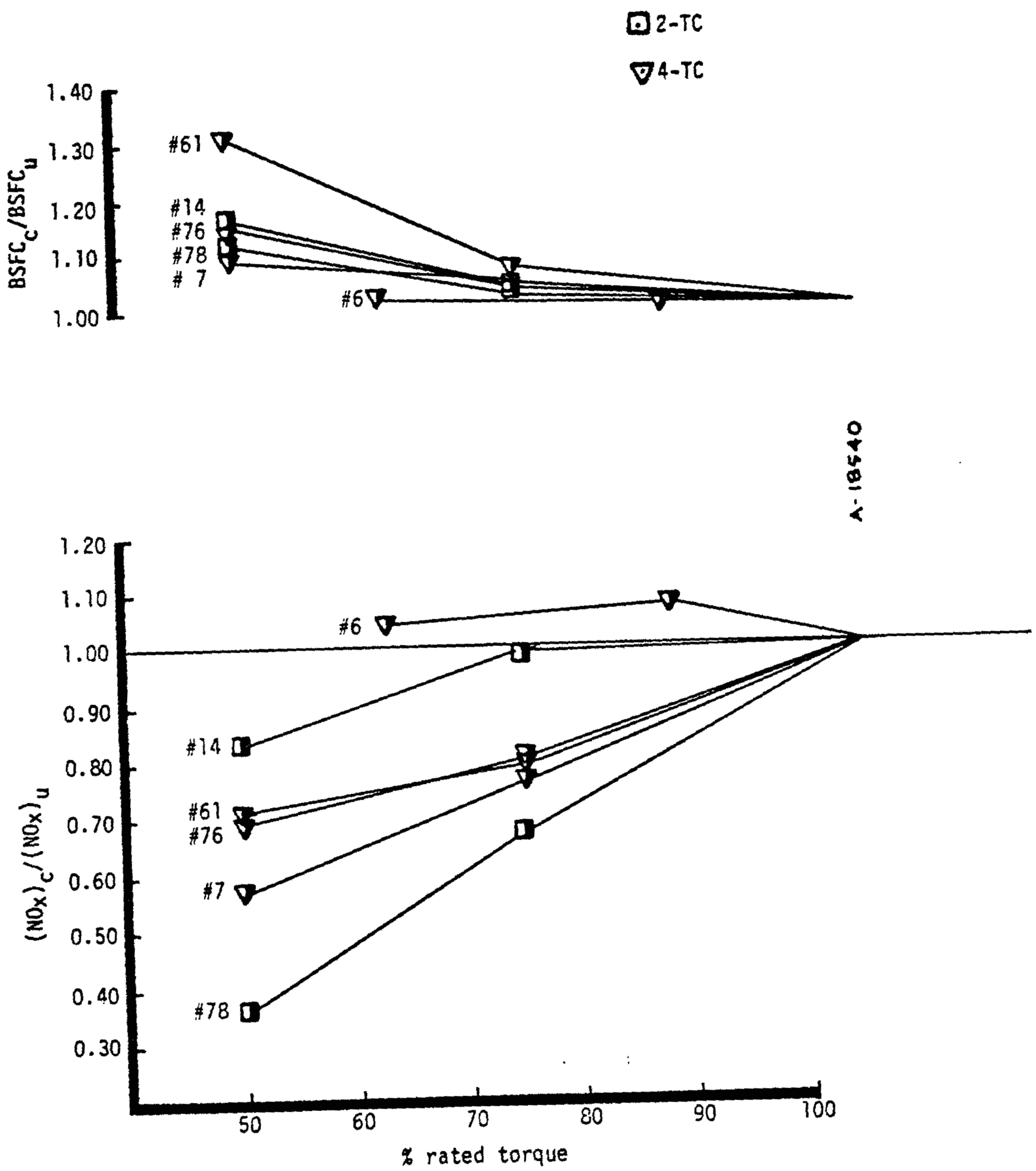


Figure 4-25(b). Effect of different amounts of derate on NO_x emissions and fuel consumption on dual fuel engines.

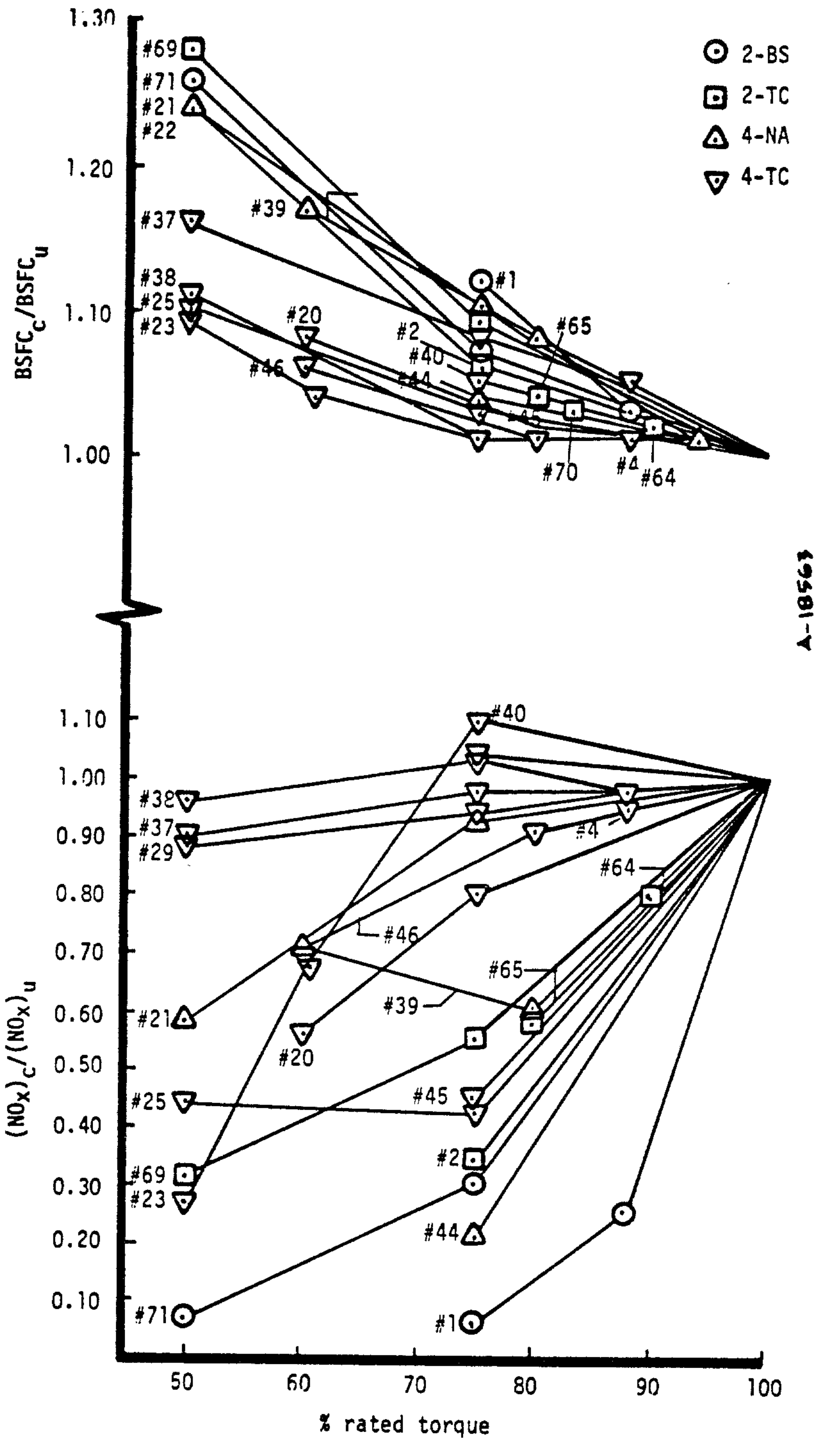


Figure 4-25(c). Effect of different amounts of derate on NO_x emissions and fuel consumption for gas engines.

Figure 4-25(b) also shows mixed results for derating dual-fuel engines but with substantially less variation than among the diesels. Thus derating caused an increase in NO_x in only one engine. In general, 25 percent derating reduces NO_x emissions 20 to 35 percent and increases fuel consumption 2 to 8 percent. Derating by 50 percent produces NO_x reductions of 30 to 65 percent, but at the same time fuel usage goes up 10 to 30 percent. In general, small amounts (25 percent or less) of derating appear effective in reducing NO_x emissions from dual fuel units, and such reductions are accompanied by fuel penalties of less than 8 percent.

Figure 4-25(c) shows that the derating of gas engines produces a wide range of NO_x reductions. In general, the nonturbocharged engines achieve the largest reductions, since derating has a greater effect on their air-to-fuel ratio. For example, blowers on blower-scavenged units operate at constant speed, independent of load; therefore, as the fuel flow is reduced to decrease output, the air-to-fuel ratio increases, causing a NO_x reduction. The turbocharged engines, in contrast, maintain a more nearly constant air-to-fuel ratio, and consequently, experience less of a NO_x reduction.

Derating does not require additional engine equipment, and the only operational adjustment is to the throttle or governor setting in order to restrict the engine power output. In most cases, this adjustment can be made in the field, although one could presumably equip a new engine with a fuel pump or carburetor whose maximum fuel delivery capacity corresponds to a derated condition. When derated, the engine's efficiency is reduced, and hence, the fuel consumption is increased. Moreover, when an engine is derated, a bigger, more expensive unit must be purchased to satisfy a given power requirement.

4.4.2 Retarded Ignition Timing

As mentioned in Chapter 3, combustion is initiated by the injection of fuel oil in a diesel or dual-fuel engine, or by a spark in a natural gas unit. The effect of variations in the timing of this injection or spark discharge is the same for both kinds of engines; that is the event can be described as combustion ignition in both cases. Therefore, this control technique is termed retarded ignition timing, or retard.

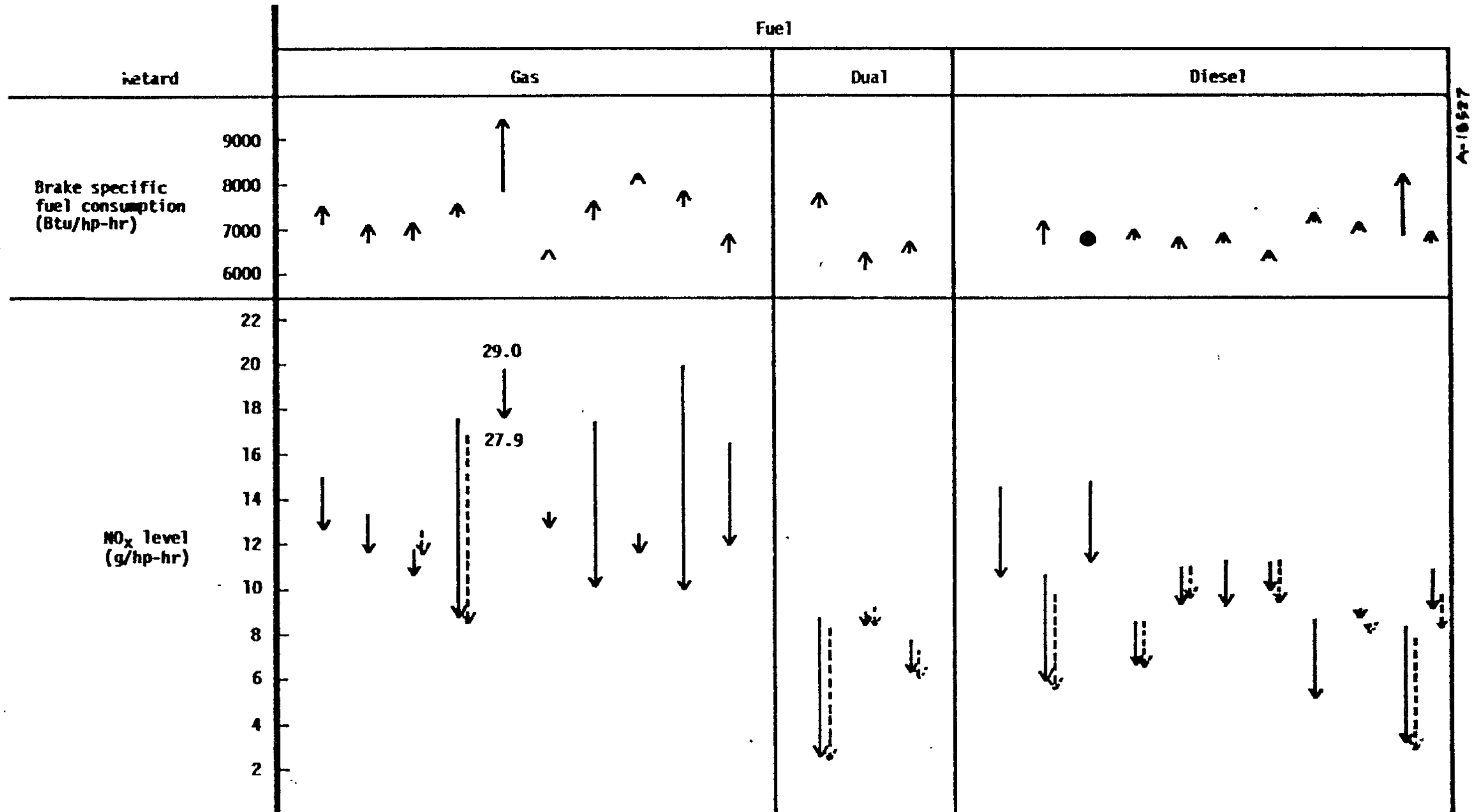
Ignition in a normally adjusted engine is set to occur shortly before the piston reaches its uppermost position (top dead center, or TDC). At TDC the air or air-fuel mixture is compressed to the maximum. The timing of the start of injection or of the spark is given in terms of the number of degrees that the crankshaft must still rotate between this event and the arrival of the piston at TDC. The extent of retard is then expressed in degrees relative to normal ignition. Typical retard values are 2° to 6°, depending on the engine. Beyond these levels fuel consumption increases rapidly, power drops, misfiring (erratic ignition) occurs, and smoke from diesel engines becomes excessive(77).

After ignition, the burning combustion gases expand, driving the piston downward. This is called the power stroke. When ignition is retarded, the duration of the combustion process does not change significantly, but rather is initiated closer to TDC and is extended longer into the power stroke. Consequently, the combustion process occurs later during higher exhaust temperatures.

In theory the fuel delivery system in diesel engines could be altered to reduce the duration of injection and thereby decrease the quantity of fuel that is combusted late in the power stroke. Such changes would require increases in the injection pressures above current levels, which are already high. One manufacturer of medium-bore engines, Cummins Engine Co., developed their own high pressure fuel pump for this application because there was no suitable commercial component on the market. Low-volume manufacturers of medium-bore engines, however, depend on outside sources for their fuel pumps and have stated that the inavailability of this critical component restricts their use of retard (e.g., to 2° to 6°) at this time[78]. One manufacturer of large-bore engines reported an unsuccessful attempt at increasing injection pressures[79]. He found that this higher pressure compressed the puel and expanded the fuel lines and consequently, the fuel injection time was not decreased. Presumably, the test could be conducted with fuel handling equipment so that expansion of the fuel lines, at least, would not prevent a manufacturer from reducing his injection period.

Retarding ignition decreases NO_x formation at the expense of reduced efficiency, thus increasing fuel consumption. Emissions of HC and CO are generally insensitive to retard except in the extreme case where misfiring can occur. That is, the higher exhaust temperatures, which tend to improve the oxidation of any remaining unburned fuel or carbon monoxide, offset the effect of shorter residence times in the cylinder. Smoke in diesel engines, however, increases rapidly after moderate degrees of retard (2° to 6°).

Figure 4-26 presents the level of reduction demonstrated for a range of engine types. Based on these data, the percent of NO_x reduction per degree of retard ranged from 1.2 to 6.9 for naturally aspirated or blower-scavenged engines and from 0.6 to 8.5 for turbocharged engines. Actual reductions due to retardation between 3 to 10 degrees ranged from 0.4 to 7.3 g/hp-hr for all engines. The effect of the control is to consistently reduce the level of NO_x produced, although the magnitude of the reduction can vary considerably between engine types or within an engine category.



Gas										
Normalized reduction[a]	2.6	3.1	2.1	16.5	0.4	0.3	4.1	1.1	5.0	3.9
Degrees retard from standard setting	6	4	4	3	9	10	10	6	10	7
Uncontrolled fuel consumption	7127	6775	6760	7265	7840	6374	7300	8139	7500	6500
Percent increase	5.2	4.3	4.7	3.4	15.3	1.4	4.0	0.2	3.5	5.5
Air charging	BS	TC	TC	TC	NA	TC	TC	TC	TC	TC
Strokes/cycle	2	2	2	2	4	4	4	4	4	4
Engine number	1	2	64	65	44	4	20	28	45	46

Dual			
Normalized reduction[a]	8.8	1.7	6.3
Degrees retard from standard setting	8	4	3
Uncontrolled fuel consumption	7040	6340	6464
Percent increase	10.1	1.1	2.9
Air charging	TC	TC	TC
Strokes/cycle	2	4	4
Engine number	13	6	16

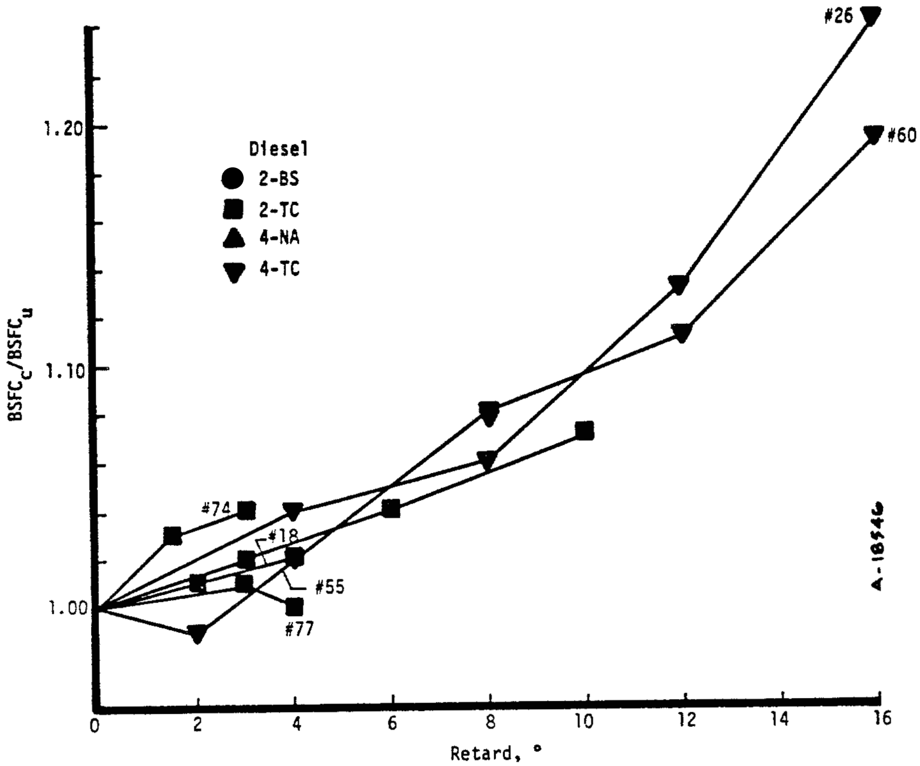
Diesel											
Normalized reduction[a]	7.0	4.4	5.7	7.1	3.8	4.4	2.7	4.9	0.7	3.8	3.9
Degrees retard from standard setting	4	10	4	3	4	4	4	8	2	16	4
Uncontrolled fuel consumption	--	6679	6734	6741	6677	6643	6366	7147	6962	6870	6643
Percent increase	--	7.4	1.9	2.5	0.8	2.6	1.5	6.4	1.6	18.7	3.6
Air charging	BS	TC	TC	TC	TC	TC	TC	TC	TC	TC	TC
Strokes/cycle	2	2	2	2	4	4	4	4	4	4	4
Engine number	17	12	18	74	5	9	15	26	55	60	8

[a]Percent NO_x reduction/degrees retard

Figure 4-26. Effect of retard on NO_X emissions and fuel consumption.

Several manufacturers have investigated the effect of different amounts of retard on NO_x emissions and fuel consumption for diesel, dual fuel, and gas engines. Their results are shown in Figures 4-27 and 4-28. Figure 4-26 shows that the effect of retarding fuel injection on NO_x levels and fuel consumption is similar for different diesel engine types. That is, 4 degrees of retard reduces NO_x from 22 to 30 percent (i.e., 26 $\pm$4 percent), and 8 degrees reduces NO_x 39 to 44 percent. Note that the NO_x reduction per degree of retard decreases for increasing levels of retard. In contrast, fuel penalties increase at a greater rate with increasing retard. Thus, 4 degrees of retard causes a 2-percent fuel penalty, 8 degrees a 6 percent penalty, and 12 degrees a 12-percent penalty. Therefore, maintenance and durability considerations aside, there are diminishing benefits to retarding diesels beyond a certain point, because increases in fuel consumption exceed decreases in NO_x levels.

Figure 4-28 shows similar results for gas and dual-fuel engines, although the data are more scattered. In general, ignition retard for gas engines is not as effective in reducing NO_x levels as it is for diesel and dual fuel engines. For example, 4 degrees of ignition retard gives about a 15 percent NO_x reduction in gas engines as compared to around 25 percent for diesel and dual fuel units. Note that the amount of NO_x reduction remains constant after a certain point for the two naturally aspirated engines, but fuel consumption continues to increase rapidly. In addition, there are practical limits of ignition retard for all gas engines. Spark-ignited engines are more sensitive to ignition timing and, therefore, misfire and exhibit poor transient performance when the ignition timing is not very close to the design point.



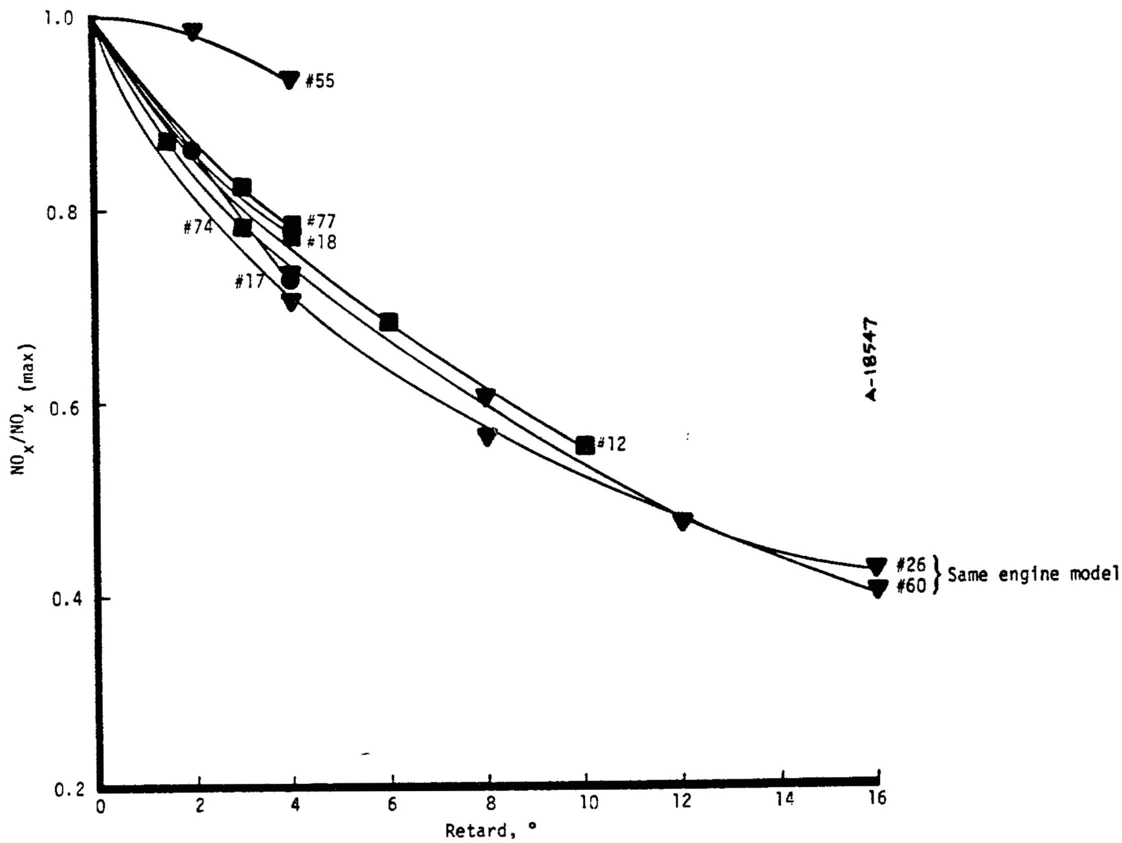


Figure 4-27. Effect of different amounts of retard on NO_x emissions and fuel consumption for diesel engines.

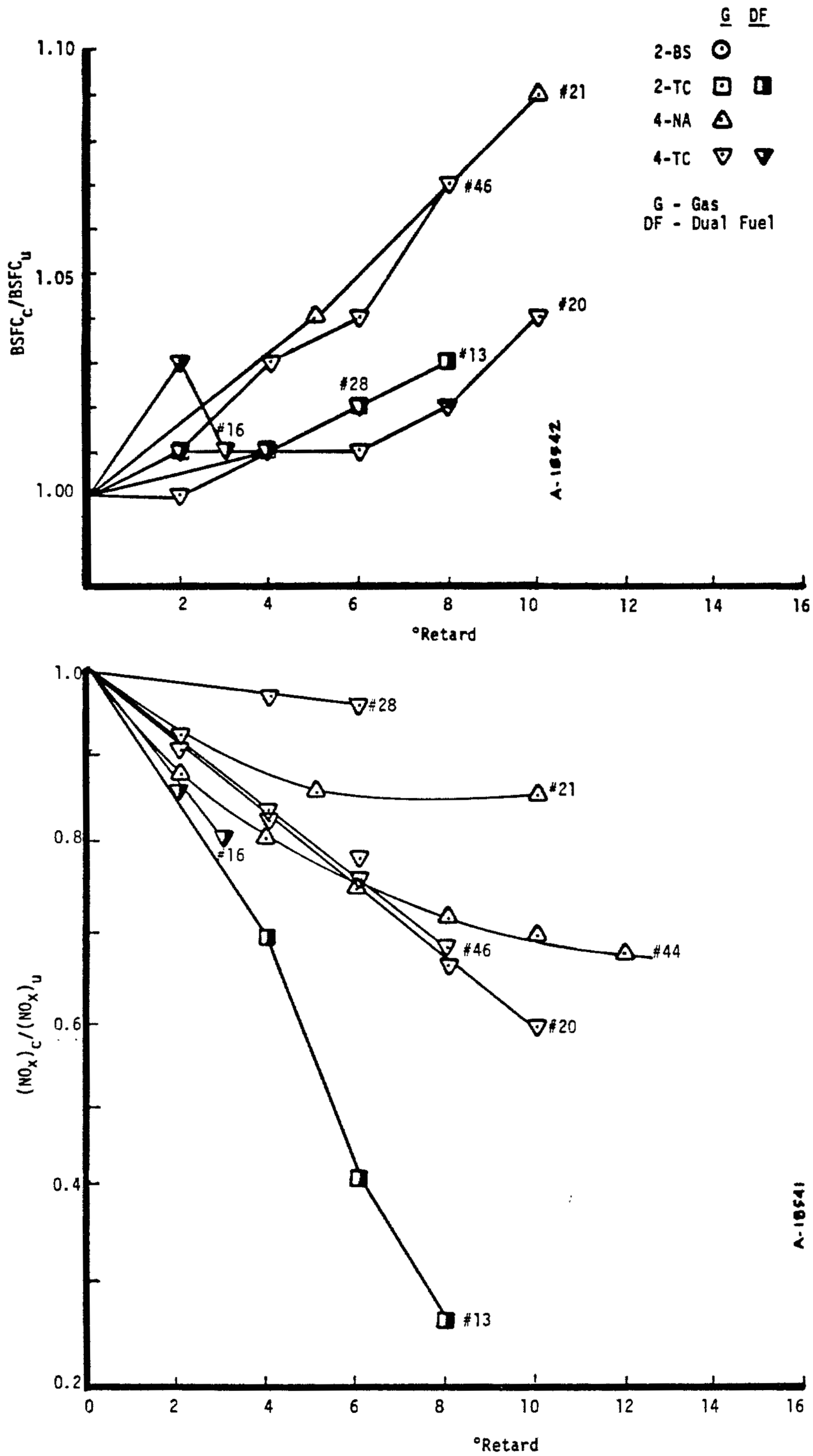


Figure 4-28. Effect of different amounts of retard on NO_X emissions and fuel consumption for gas and dual fuel engines.

Special equipment is not required for injection ignition retard as it involves only an adjustment of the engine spark or injection pump timing. Typically, the nominal setting of the ignition time is fixed by means of hardware items, such as crankshafts. Means are then provided for adjustments, or fine tuning, about this nominal value to compensate for variations in altitude, fuel, engine wear, etc. Manufacturers usually perform this fine tuning service during the production run-in of the engine, but the adjustments also can be made by the operator. This typically occurs every 10,000 hours in the course of normal maintenance, but the setting is actually verified or corrected weekly.

As stated earlier, peak cylinder temperatures and pressures are lowered by retard, and, hence, the thermal and structural loadings are lowered. However, the delayed combustion causes higher exhaust temperatures, which may lead to rapid deterioration of the exhaust valves if the exhaust temperatures exceed the design limits of the valve material. According to a manufacturer of gas-fueled engines, the values in their current production engines can withstand temperatures up to 1300°F, and the turbochargers are limited to 1200°F (these two temperature limits are not inconsistent because the exhaust gas cools between the cylinder exhaust and the turbocharger inlet)[80]. Current cylinder exhaust temperatures range from 900°F to 1250°F. Nevertheless, one manufacturer determined that 4° retard of ignition in a dual-fuel engine caused a 25-percent reduction in the maintenance life of his current valve material[81]. Another manufacturer reported that his naturally aspirated SI engines are presently operating near their exhaust material limits (1300°F) at rated load conditions. Data from one engine showed that 10 degrees of ignition retard caused the exhaust temperature to increase from 1263° to 1370°F. (NO_x emissions were reduced 17 percent.) Therefore, the application of retard to meet standards of performance may require more frequent engine maintenance or greater initial cost for higher temperature exhaust material.

4.4.3 Air-to-Fuel Ratio Changes

The air-to-fuel ratio is defined as the mass flowrate of air ingested by the engine divided by the mass flowrate of fuel consumed. This ratio is termed stoichiometric if precisely enough oxygen is present in the mixture to completely oxidize the fuel. When the ratio is greater than stoichiometric, excess air (oxygen) is present, and the mixture is referred to as lean. Conversely, a lower than stoichiometric ratio is commonly called fuel rich, or simply rich, because more fuel is present in the mixture than can be completely burned.

The maximum NO_x and minimum HC and CO emissions will generally occur at an air-to-fuel ratio slightly leaner than stoichiometric. Although maximum flame temperatures occur at less than stoichiometric ratios, maximum NO_x levels do not occur until lean A/F ratios when oxygen availability is increased. Perfect mixing of the air and the fuel never occurs in existing engines; therefore, some excess air is necessary for complete combustion and minimum HC and CO emissions. These relationships are shown in Figure 4-29(82) for a gasoline-fueled automobile engine. Similar curves apply to diesel- and gas-fired units, with different peak levels for the various curves and shifts in the air-to-fuel ratio that correspond to peak NO_x generation. When the engine is operated rich, HC emissions rise sharply because the available oxygen is no longer sufficient for complete combustion of the fuel. The lack of oxygen for combustion also means it is not present for NO_x formation and so, despite the high cylinder temperatures, NO_x formation will drop sharply at increasingly rich mixtures.

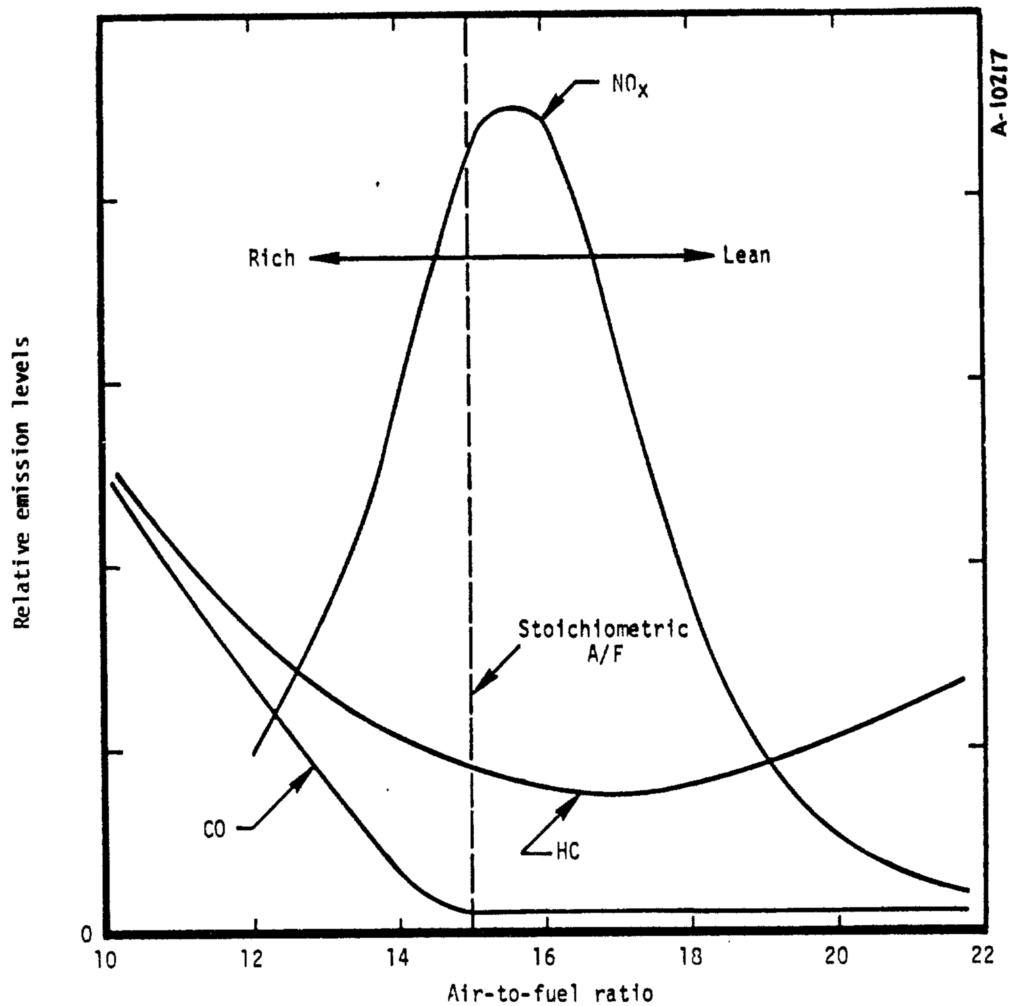


Figure 4-29. Effect of A/F ratio on emissions of a gasoline engine (Reference 82).